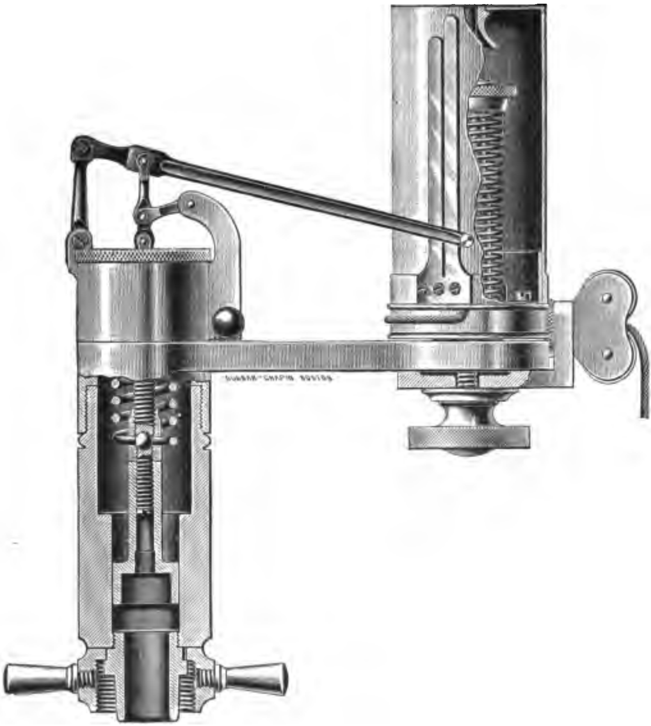
MODERN

GAS AND OIL ENGINES.



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A PRACTICAL TREATISE

ON

MODERN GAS AND OIL ENGINES,

BY

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PREFACE.

In writing these pages I have endeavoured to supply to the average mechanical draughtsman the information necessary to enable him to apply his art to the design of gas engines. A real knowledge of the elements of machines can only be acquired in the fitting and erecting shops. There the eye is trained and a sense of proportion developed which is always helpful in the design of new patterns. Unless, however, a draughtsman thoroughly understands the principles which underlie his art, he is liable to errors which prove themselves as costly as those made by the mere theorist.

In developing the conception of the work before me, I have first described the general arrangement of a gas-engine plant, then the types of modern gas engines, and have afterwards attempted to explain how their leading dimensions may be calculated.

The importance of systematic testing is now felt by many engineers. I have therefore described in detail the apparatus required and the calculations necessary to make a complete gas-engine trial. In connection with this subject a chapter is added on the practical analysis of gases.

The first part of the book concludes with a description of a series of experiments made to determine the effects of products of combustion when present in explosive mixtures of coal gas and air. Some doubt has been expressed as to the accuracy of the conclusions drawn from these experiments, on the ground that the mixing of the gases was imperfect. Having regard to the fact that at least one minute of time elapsed between filling the vessel and igniting the charge, whereas in a gas engine running at 180 revolutions per minute only one-third of a second elapses, I think it improbable that the diffusion of products of combustion in a gas-engine cylinder is more

perfect than in my experimental apparatus, notwithstanding the wide difference between the conditions.

In the second part of the book a brief description is given of the physical properties of oils, of oil-engine vaporisers, and a few special points in connection with oil-engine testing.

In conclusion, I desire to express my thanks to the following firms for their kindness in assisting me: Messrs. Crossley; Andrews; Dick, Kerr, and Co.; Tangye, Fielding, Wells Brothers, Barker, Dougill, Crosby, Globe Engineering Co., Priestman, and Elliott Brothers. I also desire to acknowledge the assistance I have derived from the works of Professor William Robinson, Professor Unwin, Mr. Bryan Donkin, Mr. Dugald Clerk, and from Professor Capper's report on the Royal Agricultural Society's Trials, held at Cambridge.

F. G.

Leeds,
July, 1897.

PREFACE TO THIRD EDITION.

THE third edition of this book has been enlarged, notably by the addition of chapters on gas engine efficiency and the application of $\theta\phi$ diagrams thereto. My experiments on the explosion pressures of acetylene and air, although published in pamphlet form, are here included; also the results of tests I have made at various times.

I am indebted to Professor Goodman and to the Council of the Yorkshire College for facilities in carrying out several experiments herein recorded during my connection with the College; also to Mr. Dent Priestman, Mr. Hugh Campbell, and others for their kind assistance.

Leeds,
February, 1902.

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MODERN GAS ENGINES.

CHAPTER I.

INTRODUCTION.

IN writing upon the subject of modern gas engines, it may at first be thought that an historical preface is out of place; but in glancing through the records of progress made during the last 35 years, it is remarkable how many of the improvements we are apt to regard as new are merely the embodiment of suggestions made by those engaged in bringing the gas engine into practical use. In more than one instance has an important advance been entirely lost sight of until, by more recent investigation, it has been again introduced and submitted to practical tests. These remarks indicate the importance of a concise knowledge of the failures and successes of early inventors, and, if need be, justify the introduction of historical matter.

It is probable that ever since gunpowder has existed inventors have sought to utilise its potential energy for the use as well as for the destruction of their fellow-men. In support of this assumption, we find records of attempts to explode gunpowder in a cylinder as early as 1678. A practical form of engine might have been evolved had it not been for the mechanical difficulties to be overcome in obtaining continuous and regular explosions with such a substance as gunpowder. It is not surprising, therefore, that at a time when the steam engine was beginning to occupy men's minds, the combustion engine was entirely

neglected; nor is it surprising that so long as an explosive substance could only readily be obtained in the form of solid matter, mechanical difficulties impeded all progress. When, however, Dr. Watson discovered that gas could be distilled from coal, and when, in the year 1792, Murdoch, a Cornish engineer, demonstrated the practical application of coal gas for lighting purposes, then we find the internal combustion engine again the subject of experiment. About this time, it will be remembered, the steam engine had become a powerful assistant in the mining industry, and was undergoing important development, brought about by the world-famed Watt. Naturally the majority of mechanicians were absorbed in furthering its possibilities. Nevertheless, from the year 1791 to 1801 three patents are on record setting forth the use of explosive mixtures of coal gas and air in motors. The first, by John Barber, explains how a wheel, with vanes upon its circumference, may be driven by releasing the pressure of an explosion through an orifice placed in proximity to the vanes. The second patent, taken out by Street in 1794, mentions the use of cylinder and piston, the latter being driven outwards by the pressure of the explosion. Flame ignition is first made use of in this engine. In 1801 another patent appeared under the name of Lebon, setting forth the advantages of compressing the gas and air before entering the explosion cylinder. Thus we see that as early as 1801 two of the principles had been explained, upon the merits of which later engines became a practical success. It appears curious that so much time should elapse between the publication of these specifications and the production of a really useful gas motor; but it is highly probable that the two factors contributing to this end were, firstly, the crude mechanical appliances for manufacturing purposes; and secondly, the tendency to follow the details of small steam-engine design, which in the matter of valves, glands, and pistons are entirely unsuitable for the peculiar conditions of gas engine work.

It is curious to note how the phases of construction of the steam engine, during a period extending over nearly 200 years, have been practically repeated in the case of the gas engine in about half the time. Just as in the steam engine, when low-pressure steam was condensed in the cylinder so that unbalanced atmospheric pressure could be utilised, so in the gas engine many attempts were made—and these were not without a measure of success—to utilise the unbalanced atmospheric pressure due to the contraction in volume of an explosive mixture, rather than the pressure produced during the explosion. In 1823 an English patent was taken out by Brown, giving the details of such an atmospheric engine. The pressure of the explosion was permitted to escape through valves in the piston, which, however, closed when the pressure of the atmosphere exceeded that of the gas. A little water injected into the cylinder assisted in cooling the products of combustion, and so reduced the pressure rapidly below that of the atmosphere. A few of these engines were constructed for commercial purposes, but owing to the difficulty in keeping them going, and to the expense in gas, they were finally abandoned. From 1823 to 1838 but little was done to popularise the gas motor, although during that lapse of time the water jacket was added to the cylinder for cooling purposes, and some attempts were made to govern the speed of an engine by controlling the admission of gas.

In 1838 Barnett made practical use of the principle of compression by constructing a single-acting motor cylinder, at each side of which was placed a gas and air pump. The three pistons moved simultaneously in the same directions; the gas and air pumps delivered into a receiver, from which communication was made to the motor cylinder as the charge was ignited. Barnett afterwards constructed double-acting engines, but with less success.

Between the years 1838 and 1860 nothing of much practical value was added to the gas motor. There were a considerable number of patents registered during this period, describing

engines working with oxygen and hydrogen gases, but no real progress was made until 1860. About this time it became recognised that double-acting cylinders were not suitable for gas engines, on account of the necessary absence of compression, the intensity of heat, the difficulties of packing and keeping a narrow piston in order. All these points were fully exemplified by the working of the Lenoir engine, which made its appearance about 1860. Although exhibiting those weak points of design, the Lenoir engine, owing to commercial agencies, became popular. The details of construction resemble somewhat those of an ordinary steam engine: a slide valve, S ports for admission and escape of the gases, electric ignition, water-jacketed cylinder and covers, were amongst the special features of the engine, Notwithstanding the consumption of gas amounted to about 100 cubic feet per horse power per hour (a fact probably not known to purchasers), this engine had a large sale. The defects of working ruined the future of the engine; and although it was subject to alteration in the hands of Hugon by the introduction of flame instead of electric ignition, it never recovered the reaction of opinion set up by public disappointment.

In 1843 Joule determined the mechanical equivalent of heat, and from that time attention was drawn to the question of thermal efficiency of motors, although it was some years before this treatment of the subject was thoroughly understood. In 1860 Barsanti and Matteucci, appreciating the convertibility of heat into motion, pointed out the necessity of rapid expansion after explosion, in order that the heat might be utilised in doing work, instead of the greater part being transmitted through the walls of the cylinder to waste itself in heating the jacket water. The engine constructed by Barsanti and Matteucci never got beyond the experimental stage, in spite of the correct principle upon which it was worked. In 1863 Otto and Langen, having also conceived the necessity for rapid expansion, devised a motor somewhat similar to Barsanti's,

making use of a free piston, which was shot upwards in a vertical cylinder by the force of the explosion. In its upward passage the weight of the piston only was overcome, but on its downward stroke the piston rod, made in the form of a rack, and geared to a pinion on the flywheel axle, caused the latter to rotate. A special ratchet gear was fitted to the pinion, so that it could revolve freely upon the flywheel axle during the upstroke of the rack. A few engines of this type are working at the present time, though they are regarded as curiosities amongst gas motors. The consumption of gas was only about 40 cubic feet per brake horse power per hour, but the noise and vibration were sufficiently objectionable features to prevent its gaining great popularity.

About this period (1861) it was again pointed out by Gustave Schmidt and Million that economy would be effected by compression before explosion. Million attempted the practical illustration of the principle of compression first upon lines similar to those of Barnard in 1838, but afterwards he introduced the compressed gas into the cylinder at the dead point, providing for that purpose an equivalent to the clearance space of a gas engine cylinder of the present day.

In 1862, a French engineer, Beau de Rochas stated definitely the conditions necessary for maximum efficiency, and formulated the well-known cycle, now termed the Otto cycle because carried into practical effect by one of that name. In the wording of the French patent taken out in 1862, Beau de Rochas manifests a clear and advanced knowledge of his subject. He says (translated): There are four conditions for perfectly utilising the force of expansion of gas in an engine :—

I. The largest possible cylinder volume contained by a minimum of surface.

II. The highest possible speed of working.

III. Maximum expansion.

IV. Maximum pressure at the beginning of expansion.

In connection with his first-stated condition, he goes on to say that a cylinder of the largest possible diameter for a given expenditure of gas should prove the most economical, and he deduces from this that only one explosion cylinder should be used in each engine. Following this significant conclusion, he touches upon the effect of time of expansion in causing the transmission of heat to the jacket water, and states definitely that the more rapid the piston speed, and consequently the expansion, the less will be the loss of heat to the jacket water. Speaking of maximum expansion, he points out that the lowest limit of pressure is soon attained, because of the rapid condensation of the gases. To obtain a maximum expansion, therefore, the gases should be compressed when cool to raise the initial pressure of explosion. In connection with this he also points out that the obvious limit to compression must occur when spontaneous combustion takes place.

Steam-engine makers, in taking up the manufacture of gas engines, have a tendency to think that the best results are only possible by a system of compounding gas engines. It is, indeed, natural that such should be the case. It should be remembered, however, that the best modern engines are working upon the precise principles laid down in 1862 by Beau de Rochas, and that their success testifies to the soundness of his views. Economy cannot be obtained by compounding, inasmuch as a great increase in compression is limited by spontaneous combustion, and expansion in two cylinders necessitates an increase in surface. Beau de Rochas anticipated in his specification still more, for he advocated combustion by compression, instead of by the electric spark, and finally gives a hint as to the value of a timing valve.

At this juncture, perhaps, the author may be permitted a digression, to arrest the attention of those apt to scoff at the value of scientific work in connection with the advancement of practical engineering. The position of science in relation to the development of the steam engine has been

ably defended from the attacks of practical men by Prof. Unwin in his "James Forrest" lecture, published in volume cxxii. of the Proceedings of the Institution of Civil Engineers. What stronger evidence than the history of the gas engine could be adduced to prove that its ultimate success depended upon the scientific principles laid down by Beau de Rochas, who, although not concerning himself with the practical details of a gas engine, so thoroughly understood the scientific side of the question. Is it not rather a reflection upon the so-called practical engineer that it took fifteen years to bring about anything like a practical development of the principles enumerated in 1862?

The Otto and Langen engine was the most practical form of gas motor in the market from 1863 to 1876. Many attempts were made to improve it, but none were effectual until Otto himself, in 1876, brought out an entirely new design of the type now familiarly associated with his name. This was the first successful engine—designed by one of high scientific attainments—in which compression took place in the explosion cylinder; and its production marks the period at which the gas motor became a serious competitor against small steam engines.

CHAPTER II.

ARRANGEMENT OF ENGINE-ROOM.

THE reader being now acquainted with the main points in connection with the historical side of the subject, we may proceed at once to a description of the details of an Otto gas engine of the type brought out in 1876, many of which are working at the present time. Although Otto engines of later design exhibit many important improvements in the details—which will be described on another page — we shall not get far astray in accepting the arrangement of gas

connections, jacket water circulation tanks, &c., as applicable to any gas engine at present on the market. On account of the unique position of the Otto engine as the pioneer of all practical gas motors, and because nearly all more modern engines work upon the Otto cycle, it is necessary for the sake of completeness to treat of it here.

The general outside appearance of the engine is somewhat similar to that of a small horizontal steam engine with overhanging cylinder, and a trunk piston, instead of piston rod and guides. Fig. 1 is a sectional plan. The piston B is shown at the back of its stroke; the space C is the clearance, provided in all gas engines, in which the charge is compressed before ignition takes place. In this engine the clearance volume is about 50 per cent of the volume swept out by the piston. The passage I is for the admission of air and gas, and also serves for the passage of flame in igniting the charge.

The burnt gases are discharged through the valve D, lifted by means of a lever worked by a cam on the lay shaft P. The slide M works to and fro between the fixed part N and the back of the cylinder, and effects both the admission of air and gas and the igniting of the charge. The slide M in moving upwards causes communication between the air passage F, the gas port G, and the entrance to the cylinder I. Air and gas are thus drawn into the cylinder upon the outstroke of the piston. The spacing of the port edges is so arranged as to allow the gas to enter the cylinder just after the air port is shut. In this position the port L is filled with gas at atmospheric pressure, which is ignited by momentary contact with flame. The slide M then moves rapidly downwards; meanwhile the piston B is returning to the back of the cylinder and compressing the charge to about 40 lb. per square inch ready for explosion. It is evident that the burning gas entrapped in the port L would be extinguished by the pressure of gas in C the moment L communicates with I, and ignition would not occur. In order to raise the pressure in the port L on its way to the

inlet I, there is a very small passage provided from the clearance space communicating with the port L before it reaches I. Through this passage the compressed gas finds its way in very small quantities into L, raises the pressure in L, and maintains the combustion by the additional supply of air and gas. The passage through which this communication is made is too small to permit the firing of the charge in the cylinder, and this does not take place until the port L—carrying its burning gas now at the same pressure as that in C—reaches I and ignites the charge.

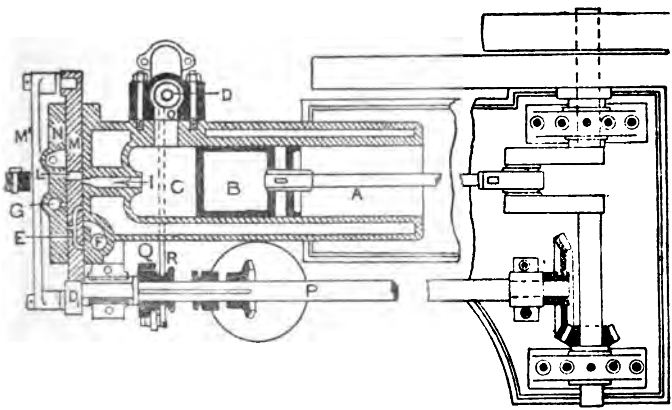


FIG. 1.—Sectional plan of Otto gas engine.

The piston is now driven forward by the explosion. The flywheels carry the piston back, driving the burnt gases out through D. It will be observed that an impulse is only given every two revolutions of the engine; hence the necessity for exceedingly heavy flywheels to maintain a constant speed. The Otto cycle, upon which nearly all modern engines work, may be summed up as follows :—

First revolution... { Outstroke draws in air and gas.
Instroke compresses charge.

Second revolution { Outstroke caused by the explosion.
Instroke discharges burnt products.

In the description of various designs of gas engines, we shall afterwards see that the mechanism for effecting this cycle of operations has undergone many changes. For the present we may turn our attention to the arrangement of the engine-room best suited for the working of a gas engine, and to those fittings external to the engine itself. When possible, the room in which a *large* gas engine is placed should be quite separated from any building in which vibration is likely to cause annoyance. If, however, it is not possible to provide a separate engine-house, two precautions may be taken to prevent unpleasant results. There should be no communication between the main building and the engine-room, excepting by means of well-fitting doors, which should be made to close automatically. This caution is not necessary in the case of small engines, but large engines of from 40 to 60 I.H.P., running at a speed of from 150 to 200 revolutions per minute, cause a rapid pulsation of air owing to the displacement of the large trunk piston; this pulsation may be transmitted through the passages and corridor to almost every room in the building, causing even the windows to vibrate to every revolution of the engine. For the same reason the engine-room should not be ventilated by connection with the air shafts of the building. Artificial ventilation is absolutely necessary, as the engine-room otherwise becomes unbearable. A small air propeller driven from the engine will answer this purpose admirably.

Another fruitful source of annoyance is the vibration of the engine transmitted through its foundations to the walls of the building. In some instances it has been found that an increase in the speed has checked the vibration of the foundations and walls. It is quite probable that every building has a period of vibration peculiar to itself, and if, therefore, a machine is running so as to have the same period of vibration, the results may be very noticeable. Increase the speed of the machine, alter its period of vibration until it no longer synchronises with that of the building, and the latter will not respond; thus the effect will be minimised to an inappreciable extent.

Another method by which vibration of the engine itself may be prevented from transmission to the building is to bolt the engine bed down upon a somewhat soft and springy material. Let some timber baulks be placed longitudinally with the engine bed. Over these put a layer of iron plate, say $\frac{3}{8}$ in. thick, then put over the plate a layer of cocoanut matting. It is well to have these mats specially made. They should be manufactured after the manner of a thick pile cocoanut door mat; two should be made of the required area; the pile faces should be put together, and the two bound at the edges into one thick mat. By putting them together in this way a thickness of 4 in. or 5 in. may be obtained of springy texture, and presenting an exterior of comparatively tough material. Another plate over the mat serves to distribute the weight of the engine over the entire area, and then the whole may be loosely held together by means of the foundation bolts. If this method is adopted, the engine will no doubt rock considerably, and extra large bolts may be necessary for holding down. The rock of the engine will cause considerable oscillation of the belt, and for this reason a rather short drive is to be preferred. These objections are, however, entirely outweighed by the satisfactory prevention of the transmission of vibration throughout the building.

The next point to be considered in the arrangement of the engine-room is the position and size of the gas meter required. All engines should be provided with a separate gas meter, so that the consumption of the engine may be checked independently. The meter should be placed outside the engine-room in an atmosphere of normal temperature, for it must not be forgotten that an increase of temperature results in an increase in the volume of gas. If, therefore, the meter is kept at a high temperature, the cubic feet registered by it will be greater for the same weight of gas than if kept at normal atmospheric temperature. No injustice is done to the producers of the gas by taking this precaution, because the temperature of the supply is itself normal.

The size of a gas meter is usually gauged by the maximum number of lights for which it is designed. Thus, small houses are fitted with a 5-light meter, others a 10-light meter, according to the requirements, and this indication of the size is usually printed upon the index of the meter. A useful rule for determining the size of a dry meter required for a gas engine, the brake horse power of which is known, is as follows: $3{\cdot}4 \times$ brake horse power $+ 5 =$ size of meter required (measured as above stated).

Thus, suppose an engine of 30 brake horse power is being erected, the meter required will be $3{\cdot}4 \times 30 + 5 = 107$-light meter. In this instance a 100-light meter would suffice.

Gas and oil engine makers are now quoting in their catalogues the brake or effective horse power, instead of indicated and nominal horse powers. The relation between these will be dealt with upon following pages, devoted to gas-engine testing; but it may here be said that this innovation is a great improvement, inasmuch as the purchaser knows exactly the capacity of an engine for doing useful work.

Having determined the size of meter suitable for measuring the gas supplied to an engine, we must next consider the size of pipe from the meter to the engine. The following rule gives the correct size of pipe for a given size of meter:

Meter size (measured in lights) $\times 0{\cdot}008 + 0{\cdot}75 =$ bore of pipe in inches. In the example previously worked it was calculated that a 100-light meter is suitable for a 30 brake horse power engine. The size of gas pipe for this meter will be equal to $100 \times 0{\cdot}008 + 0{\cdot}75 = 0{\cdot}8 + 0{\cdot}75 = 1{\cdot}55$ in. Putting in a size of pipe to the nearest $\frac{1}{8}$ in., we should, in this instance, couple the gas supply to the engine by means of a $1\frac{1}{2}$ in. diameter pipe.

Another rule may be used for obtaining the diameter of gas pipe when only the brake horse power is known. If D = inside diameter of pipe in inches, then we have—

$$D = 0{\cdot}027 \times \text{brake horse power} + 0{\cdot}75.$$

Working out the diameter suitable for a 30 brake horse power engine, we have—

$$D = 0{\cdot}027 \times 30 + 0{\cdot}75$$
$$= 0{\cdot}81 + 0{\cdot}75$$
$$= 1{\cdot}56 \text{ in.};$$

and putting in a size to the nearest $\frac{1}{8}$ in., we obtain $1\frac{1}{2}$ in. diameter, as before calculated from the other data.

Between the gas meter and the engine there should be fitted a flexible bag through which the gas passes. The bag may be made by clamping two sheets of indiarubber about $\frac{1}{8}$ in. thick to each side of a cast-iron ring, the diameter of which may be from 1 ft. 6 in. to 2 ft. 6 in., or more, according to circumstances. By this arrangement rapid fluctuation of pressure in the gas mains is prevented, thereby maintaining a steady light given by the burners in proximity to the engine. To contribute to this end, two bags are sometimes fitted in the following way: The bags, each composed of a cast-iron dish to form one side, and sheet rubber the other, are connected together, so that the gas enters and fills one, then passes on to the next. Between the entrance to the first gas bag and its pipe flange is fitted a cast-iron box containing a plain slide valve. This valve is moved by a lever attached to the centre of the rubber side of the bag in such a way as to close the entrance to the bag when the latter is full of gas. A similar valve is fitted between the two consecutive bags, and the whole forms a very efficient means of governing the pressure in the mains. In putting up the gas piping and bags, the rubber sides should be placed next the wall, and the bags should be as near the engine as possible. It is advisable to leave sufficient room between the wall and rubber to enable the attendant at any time to place his hand upon the rubber to flatten it. It is often necessary to do this in starting an engine, to get rid of an accumulation of poor gas in the bags when the machinery has been standing several days. It is also a useful precaution to have a small gas cock fitted to the mains near the meter, to which a U glass tube may be attached by an indiarubber pipe

By partially filling the tube with water and opening the cock the fluctuation of pressure in the mains may be observed, and the efficiency of the gas bags tested when the engine is running. The rubber side of the bags must be kept perfectly clean and free from oil, or its corrosive effect will soon be evident, and, further, the rubber must not be subject to excessive temperatures.

The diameter of the engine exhaust pipe may be found from the following rule for any engine larger than 5 brake horse power: If D_e = diameter of exhaust pipe,

then we have $D_e = 0{\cdot}528$ horse power $^{0{\cdot}57}$

Taking the brake horse power at 30, as in the previous example, and applying the rule given to find the exhaust diameter, we have—

$$0{\cdot}528 \times 30^{0{\cdot}57} = D_e,$$

$$\log 30 = 1{\cdot}477\ ;^*$$

and $1{\cdot}477 \times 0{\cdot}57 = 0{\cdot}838$ (nearly) = the log of $(30^{0{\cdot}57})$.

Finding the number corresponding to 0·838, we obtain 6·89 and $0{\cdot}528 \times 6{\cdot}89 = 3{\cdot}64$ diameter of exhaust pipe in *inches*.

We should here adopt a 3⅝ in. or 3¾ in. diameter pipe. Engines of from 1 to 5 brake horse power may have the exhaust pipes from 1 in. diameter to 1¾ in.

It is very important that the exhaust pipe should be free from bends, but where necessary they should be of large radius, say 6 in. Owing to the high temperature of the exhaust gases, the pipe should be isolated from inflammable material. When the mouth of the pipe discharges into the open air it should point downwards, to prevent the collection of rain water. When it is desirable to deaden the noise of the exhaust, it may be discharged into a cast-iron chamber partially filled with flint pebbles, or into the bottom of a trench filled with pebbles. Whatever form of

* All calculations will be worked out upon a 10 in. slide rule, and the results will be accurate to about one-half per cent.

exhaust chamber is made use of, it should always be placed in an accessible place, and provision made for efficiently draining water which may collect.

The next subject for consideration in the arrangement of gas-engine plant is the delivery of water to the jacket surrounding the working cylinder of the engine. The jacket water circulates round the cylinder in an annular space, formed by the outer wall of the explosion cylinder and the inner wall of a concentric casing. This casing is cast with the cylinder, and the space allowed for the water is from $\frac{3}{4}$ in. to 2 in. A flange is provided at the under side of the jacket for the inlet of the water, and another at the top and front end for the outlet. For small engines the inlet and outlet pipes may be the same size, but when circulation tanks are used, and the engine is over 20 horse power, it is better to have the outlet pipe from $\frac{1}{2}$ in. to $\frac{3}{4}$ in. larger in diameter.

The function of the jacket water is to carry away the excess of heat due to combustion in the cylinder, which, under the present conditions, cannot be converted into work. It is found in practice that the quantity of heat accounted for by the jacket water amounts to from 30 to 50 per cent of the total heat derived from the combustion of the gases in the cylinder, and that under these conditions lubrication is possible. In estimating, therefore, the quantity of jacket water required, it is necessary to bear in mind that it must absorb not less than 30 per cent of the heat of combustion, and that its maximum temperature should not exceed 150 deg. Fah. for continuous running. When the cooling water runs to waste, the maximum temperature may be somewhat higher, but when circulating tanks are used it is better not to exceed 150 deg. Fah. If we assume an average gas consumption of 22 cubic feet per indicated horse power per hour, we may estimate the heat units generated in the cylinder for each H.P. to be $22 \times 620 = 13{,}640$ units per hour, because 1 cubic foot of coal gas develops on an average 620 British thermal units. At our

lowest estimate, 30 per cent of this must be taken up by the jacket water; this amounts to 4,092 units. We have said that the jacket water should not exceed 150 deg. Its inlet temperature may be taken at 60 deg. Fah., thus allowing a rise of temperature of 150 − 60 = 90 deg. Fah. per pound of water, the pounds per I.H.P. required $= \frac{4092}{90} = 45{\cdot}5$ (nearly) per hour. This is equivalent to $4\frac{1}{2}$ gallons of water per I.H.P. per hour, and it approaches the minimum quantity allowable for continuous working. Thus we find that if the water supply is continuous, and passes through the jacket, afterwards running to waste, at the rate of $4\frac{1}{2}$ gallons per I.H.P. per hour, efficient lubrication may be maintained. With such an arrangement, the size of the jacket inlet and outlet pipes will depend entirely upon the head of water in the source of supply, but as the arrangement described is only suitable for experimental engines and other special cases, the jacket pipes are put in from 1 in. to 2 in. diameter up to engines of 20 I.H.P. Above 20 H.P. the diameters range from 2 in. to 3 in. for the inlet, and from $2\frac{1}{2}$ in. to $3\frac{3}{4}$ in. for the outlet.

The most usual method of supplying the jacket is to place it in communication with a set of tanks, through which there is a free circulation of water. The diagram (fig. 2) shows an arrangement of circulation tanks. The tank A is coupled to the jacket inlet, and receives the feed necessary to make up the loss from evaporation. The outlet pipe is taken to the normal water level of the tanks. This pipe should have a minimum rise of 2 in. per foot where it leads into the tank, in order to secure an easy circulation. The tank B communicates with A through a pipe starting from the bottom of B, passing out at the normal water level across to A. A drain pipe, slightly above the normal water level, serves to take away any excess of feed water. For clearness in the drawing the engine is shown between the tanks, but it is usual to have the tanks side by side, outside the engine-room, in as cool a place as possible. They should,

however, not be placed in the open air, as, in the event of severe frost, much inconvenience may be occasioned. In this connection it may be well to mention, that when an engine is subject to frost, and is not running continuously, it is a wise precaution to have two valves fitted to the inlet jacket pipe, the one near the tank A to shut off the supply from A; the other at the bend of the pipe as it enters the jacket, and communicating with the atmosphere. Thus the jacket only may be drained, and be secured from bursting if the frost gets to the engine. The circulation tanks should have a capacity of from 20 gallons to 30 gallons per I.H.P.,

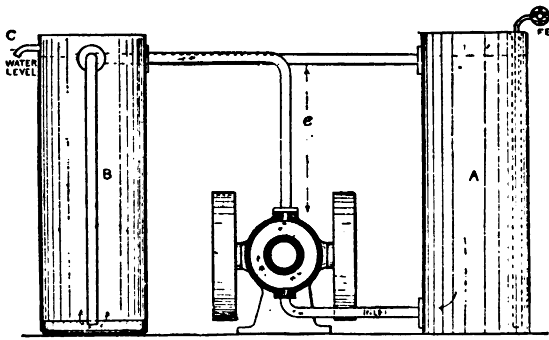


FIG. 2.—Arrangement of water tanks.

so as to allow a sufficient time for cooling. The form of tanks is usually cylindrical, though need not be. It is, however, necessary that the effective height should be from two and a half to three times the diameter or breadth, otherwise a good circulation will not be induced.

The circulation is induced by the expansion of the column of water in the pipe *e* leading from the jacket to the tank. When the engine is started the jacket temperature may reach about 160 deg. Fah., and the volume of the water will increase about $\frac{1}{30}$th. Thus, a column of water 8ft. high will increase when heated nearly to 8ft. $3\frac{1}{2}$ in. There will

then be an equivalent head of water in the cold tank A of $3\frac{1}{2}$ in., which, neglecting friction in the pipes, will cause the cold water to enter the jacket at the rate of 4·4 ft. per second. Thus, immediately upon starting the engine, a very brisk circulation of the water is induced. After some time has elapsed, the water in A rises nearly to the same temperature as at the bottom of B, so that the equivalent head, which afterwards maintains the circulation, may be calculated from the difference in temperature between a stratum of water at the bottom and that at the top of the tank B.

We have now dealt with the chief external fittings of a gas engine worked by town's gas. The economy to be gained by putting down plant for the production of Dowson gas depends upon many conditions which may or may not exist in any particular instance. Undoubtedly, for the constant running of large engines, producer gas is by far the most economical. This important subject will be more fully discussed later.

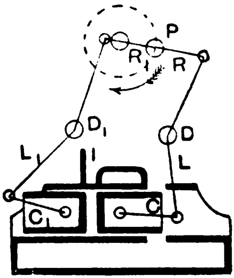
Before concluding these remarks upon the arrangement of an engine-room the following points may be noted: When possible, arrange for the tight side of the belt to run from the bottom of the driving-wheel. This is not always convenient when driving a dynamo, for the dynamo pulley must run in the same direction as the hands of a watch when viewed from a position facing the pulley end of the dynamo. When cramped for room, it is necessary that the gas-engine cylinder should lie between the dynamo and the engine crank shaft; thus, with an engine receiving its impulse on the top throw of the crank, the tight side will be at the top. With a fair width of belt no serious amount of slip will take place in transmitting, say, 40 horse power, even though the driving pulley be 7 ft. diameter, the driven pulley 18 in., and the shaft centres only 15 ft. apart. Gas engines have in some instances been coupled directly to the dynamo shaft, thus effecting a great saving of space. At Sunderland a gas-engine plant has recently been put down driving the centrifugal pumps at the docks without the use of either belts or gearing.

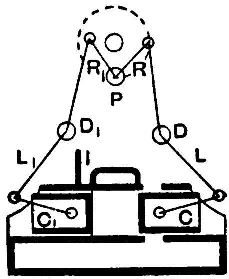
CHAPTER III.

TYPES OF GAS ENGINES.

ATKINSON'S "DIFFERENTIAL" AND "CYCLE" ENGINES.

SINCE the Otto patent expired, the commercial value of other engines has much depreciated, and in many cases makers have abandoned their own patents in favour of the Otto principle. In 1885, Mr. J. Atkinson, M.I.M.E., patented a motor known as the Differential engine. This engine was constructed upon different lines to any other gas engine that has been made, and from a theoretical point of view is the most perfect engine yet invented. Regarded however as a mechanical contrivance, Mr. Atkinson himself says it is not good, though it may be considered an ingenious combination of link motion. Although not now manufactured, no work upon the gas engine would be complete without a description of this machine. Figs. 3, 4, 5, and 6 show the engine diagrammatically in four positions of the crank pin. The cylinder is open at both ends, and water jacketed on the top only. Two pistons work in the cylinder, and are each connected with the crank pin P by means of a freely jointed connecting rod *c*, attached to the bent lever L. This lever oscillates upon the fixed pin D, which is attached to the framing of the engine. The upper end of the lever L drives the crank pin by means of the short connecting rod R. Fig. 3 shows the engine with the charge compressed between the two pistons. The left-hand piston is just uncovering the entrance to the ignition tube I. It will be observed that as the crank pin follows the direction indicated by the arrow, the rod R_1 merely revolves about the upper end of the lever L_1 until P arrives at position shown in fig. 5. The left-hand piston has therefore remained almost stationary, whilst the right-hand piston has moved very rapidly to the right

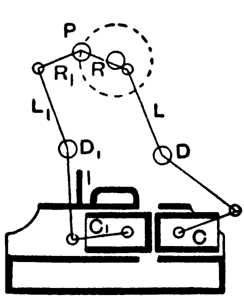


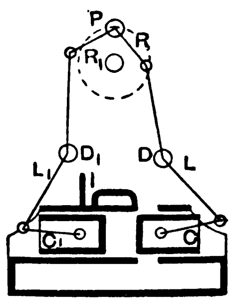


ATKINSON'S DIFFERENTIAL GAS ENGINE.

FIG. 3.—Volume before explosion.

FIG. 4.—Volume when expanded.

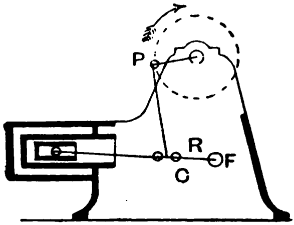


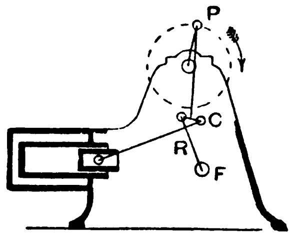


ATKINSON'S DIFFERENTIAL GAS ENGINE.

FIG. 5.—Showing total expulsion of burnt gases.

FIG. 6.—Volume of charge before compression.

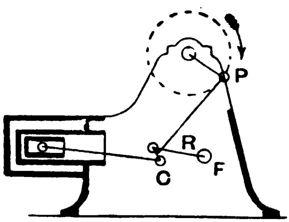


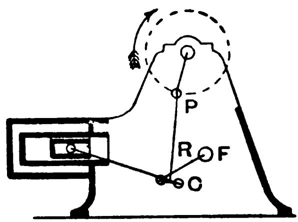


ATKINSON'S CYCLE GAS ENGINE.

FIG. 7.—Showing volume before explosion.

FIG. 8.—Showing volume when expanded.





ATKINSON'S CYCLE GAS ENGINE.

FIG. 9.—Showing total expulsion of burnt products.

FIG. 10.—Showing volume of charge before compression.

and the charge between the pistons is expanded, thus giving an impulse to the crank pin in the direction of its motion. The effect of the pressure upon the left-hand piston during the time the crank pin moves between the positions shown in figs. 3 and 4 merely produces a downward thrust upon the crank-shaft journals, and only a small resolved force acts against the rotation of the crank pin. On the other hand, the rod R, attached to the right-hand lever L, exerts an almost tangential effort on the crank pin during this quadrant. The exhaust port is now uncovered by the piston c, and remains so during the next quarter revolution. In the mean time the left-hand piston c_1 moves rapidly to the right, and expels the whole of the burnt gases. The piston c now moves *slowly* to the left, covering the exhaust port, while c_1 moves *rapidly* to the left, drawing in the fresh charge. Fig. 6 shows the position of the pistons when the charge is complete, and it should be observed that the volume enclosed by the pistons is here *less* than in fig. 4, just when the exhaust opens. It therefore follows that the charge is expanded beyond its original volume at atmospheric pressure, thus carrying out the principle of maximum expansion laid down by Beau de Rochas.

The four characteristic features of this engine are—an ignition per revolution, the expulsion of the burnt gases, expansion to greater than the original volume, also perfectly automatic valves and means of timing the ignition. Notwithstanding the mechanical defects, this engine was the most economical of its time, for it consumed only 26 cubic feet of gas per brake horse power in small engines, and this was reduced to 24 cubic feet in large engines. This engine was first exhibited at the Inventions Exhibition held in London in 1885, and was there awarded a gold medal.

Its mechanical defects were so obvious that its manufacture was abandoned in favour of another type, patented by Mr. Atkinson, and known as the Cycle engine. Here the patentee sought to combine the advantages of the Differential engine with more durable mechanism, and he was able,

by the arrangement shown in figs. 7 to 10, to reduce the gas consumption to 22 cubic feet per brake horse power per hour. Fig. 7 shows the position of the piston when ignition takes place. The fixed centre F being rather below the horizontal centre line of the cylinder causes the rod R to rise through a greater angle above than below F. By this means the explosion stroke was 11·1 in. Fig. 8 gives the position of the links at the end of the explosion stroke.

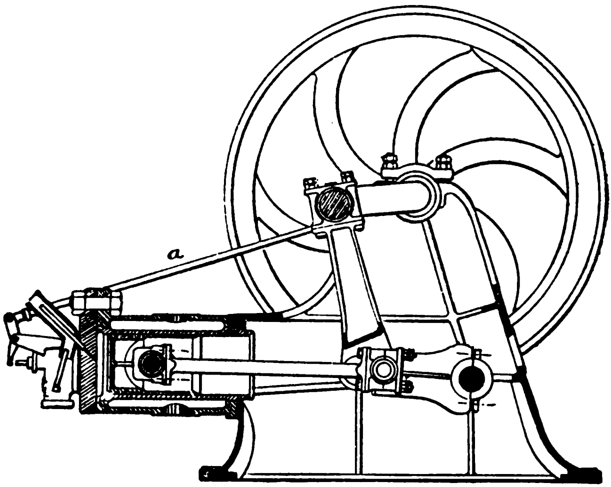


FIG. 11.—ATKINSON'S CYCLE GAS ENGINE.

During the exhaust stroke it will be noticed (fig. 9) that the joints at C are inclined instead of horizontal, as in fig. 7, thus increasing the length of the backward stroke to 12·4 in., and bringing the piston close to the back of the cylinder. The whole of the burnt gases being driven out, the piston makes another forward stroke, but, as before mentioned, owing to the smaller angle made by the link R below the horizontal, this stroke only measures 6·3 in. Fig. 10 shows the position of the crank pin when the charge is

completed. Fig. 11 gives a more complete view of the Cycle engine, many of which are at the present time working.

The rod *a* operates the gas valve by means of a cam on the crank shaft. A similar rod in front of *a* operates the exhaust valve. The ignition tube in this engine is screwed direct into the back of the cylinder, the time of ignition being determined by the length of the tube. The hot ignition tube communicates with the combustion chamber by means of the small hole, through which passage the fresh charge is compressed into the tube. After the explosion, expansion, and exhaust have occurred, the tube remains full of products of combustion, and it is evident that, before a fresh charge can be ignited, these products must be so far driven to the end of the ignition tube as to allow the fresh mixture to come in contact with the red-hot portion of the tube. Hence the length of an ignition tube plays an important part in determining the amount of compression necessary, and consequently the time of an explosion. In the Cycle engine the firing of the charge is regular, and little trouble is experienced with it. This is probably, in a great measure, due to the pure charge put into the cylinder. There is much diversity of opinion upon the value of a timing valve.

To show the result of the application by Mr. Atkinson of the principles laid down by Beau de Rochas, the following figures may be quoted. A 6 horse power Cycle engine was put down by the Hampton Wick Local Board to drive a single-acting air compressor in connection with the Shone pneumatic sewage system. This was tested against a duplicate set of pumps driven by a Crossley Otto engine under precisely similar conditions. It was found that the Cycle engine consumed 18·4 per cent less gas than the Crossley engine. Notwithstanding the high efficiency of the Cycle engine with regard to gas consumption, it has been found to be costly in up-keep, requiring more lubrication and more frequent repairing, consequently, now that the Otto patents have become public property, the Cycle engine is not manufactured. It is nevertheless worthy of

careful study as an example of a very successful attempt to put into practice the true theory of an economical gas engine.

It has been stated previously that the majority of gas engines now being manufactured work on the Otto cycle, but owing to the irregularity of the impulse given to the piston it is certainly desirable to aim at a more equal distribution of the turning effort exerted upon the crank pin. This object may be to some extent attained by building an engine with two Otto cycle cylinders instead of one, and a still greater regularity in the impulses has been attained by the construction of double-acting cylinders, made possible by improvements in details relating to lubrication and glands.

COMPARATIVE CYCLES OF GAS ENGINES.

The following diagram, fig. 12, shows the steadiness of running with the various cycles now used. The shaded squares represent indicator diagrams taken from the back or front of the piston when an explosion takes place. It is seen that in the Otto cycle single-cylinder engine, in six revolutions, only three impulses are given to the piston. The Griffin engine, about which more will be written, has a double-acting cylinder. The series of operations on one side of the piston only is as follows :—

1. Outstroke : Explosion and expansion.
2. Instroke : Products of combustion expelled.
3. Scavenger stroke—outstroke : Draws in pure air only.
4. Scavenger stroke—instroke : Clears out the cylinder.
5. Outstroke : Draws in gas and air.
6. Instroke : Compresses ready for explosion.

It is therefore evident that the back of the piston receives one impulse every three revolutions. Similarly, the front of the piston receives one impulse every three revolutions; consequently during the six revolutions two impulses are received by one side of the piston, and two by the other, making in all four. The object of the third outstroke for drawing into the cylinder a charge of pure air will be fully

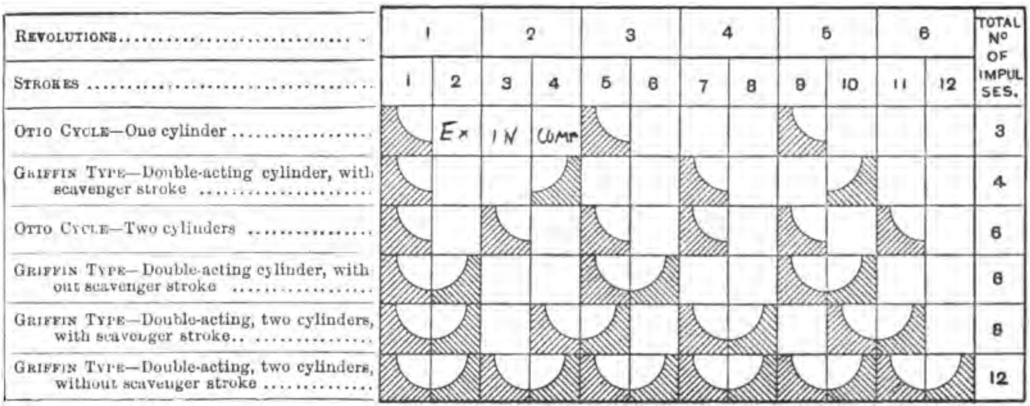


FIG. 12.—DIAGRAM SHOWING COMPARATIVE CYCLES OF GAS ENGINES RUNNING AT FULL LOAD.

treated later; but here it may be stated that if all the products of combustion are entirely swept out of the cylinder by this means, it is generally believed that the efficiency of the next charge of gas is increased. This is no doubt true when Dowson gas is used, but experiments carried out by the author upon mixtures of coal gas and air point to the conclusion that the deleterious effect of burnt products is much overrated, excepting when the gas is more than 7½ per cent of the air by volume.

The Otto cycle, when applied to an engine having two cylinders, gives an impulse every revolution of the fly-wheels in the order shown upon the diagram. The Griffin type, having but one double-acting cylinder, and working without a scavenger stroke, gives an average of one impulse every revolution, though the sequence of the working stroke is perhaps somewhat less conducive to steady running. This is, however, improved by the use of two cylinders, each with its scavenging strokes, giving eight impulses in six revolutions. And, lastly, the Griffin engine, with two double-acting cylinders and no scavenger stroke, gives twelve impulses during six revolutions. Regarding only the number and sequence of motor strokes per revolution, the above-mentioned cases afford an example of what may be done by the various combinations of cylinders; but it will be understood that there are other engines, which, whilst fulfilling one set of conditions indicated by the diagram, are distinct in design, and possess advantages peculiar to themselves.

THE CROSSLEY ENGINE.

The modern Otto, as built by Messrs. Crossley Brothers Limited, Manchester, is very different from that illustrated in fig. 1. The old method of flame ignition is now entirely superseded by allowing the explosive mixture in the cylinder to enter a red-hot tube attached to the cylinder. The slide valve controlling the admission of the air and gas has been replaced by two simple mushroom valves, somewhat similar

to the exhaust valve on the original Otto engine. There are many objections to the employment of a slide valve on a gas engine, and it is probable that a change would have been made much earlier had it not been generally believed that the slide valve effected a better distribution of the gas in the cylinder than was otherwise possible. In describing the original Otto, it was pointed out that the spacing of the port edges, admitting the air and gas, was such that air entered the cylinder first, then air and gas, and lastly pure gas only. It was believed that when the full charge had entered the cylinder it existed in three distinct layers—the purer gas assisted the ignition, the more dilute mixture spread the flame, and, lastly, the pure air deadened to some extent the percussive effect upon the piston. A deceptive experiment in support of this theory may be made by fitting a simple piston in a glass tube. Let a cork be fitted tightly to the end of the tube beneath the piston, and into a hole in the cork pass a cigarette. Push the piston to the bottom of the tube, and then withdraw it *slowly* to about half the length of the tube. Upon lighting the cigarette, and completing the slow movement of the piston to the end of the tube, smoke will follow in a distinct stratum, and will remain unmixed with the air for some time. This slow movement, however, does not exist in a gas engine, and, moreover, the gas more readily follows the piston than in our experiment, and consequently a practically homogeneous mixture is contained in the cylinder before ignition takes place. That there is no stratum of pure air next the piston of a gas engine has been proved by igniting the charge at the forward end of the cylinder, instead of at the back of the clearance space.

As the theory of stratification was proved to be a fallacy, there appeared no reason why the mechanical difficulties of the slide valve should not be overcome by the use of separate gas and air valves. By the use of mushroom valves leakage is entirely prevented, rendering it possible to compress up to about 80 lb. or more per square inch before ignition,

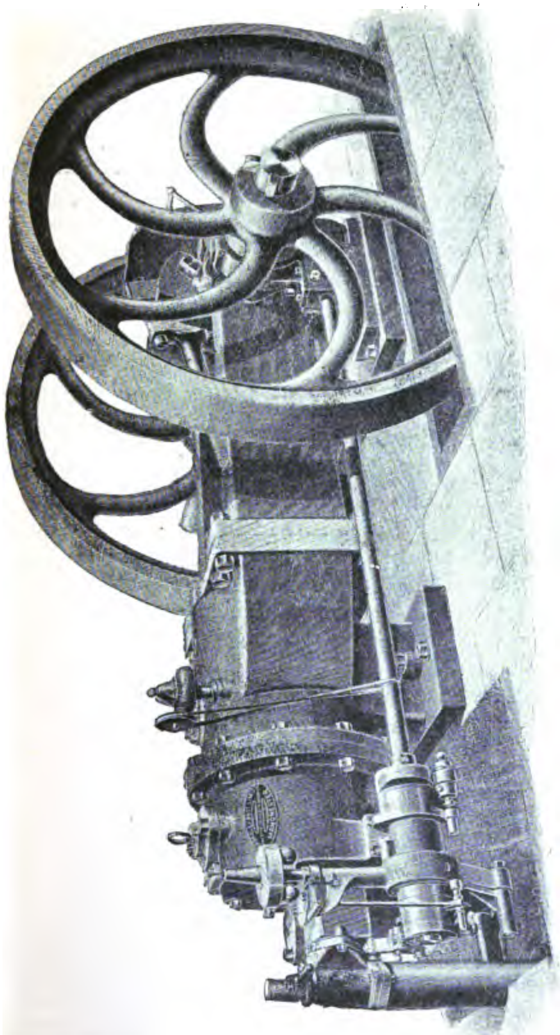


Fig. 13—CROSSLEY'S SINGLE CYLINDER GAS ENGINE.

whereas in the old slide engines a pressure of only 40 lb. was obtained. These valves are invariably held upon their seats by a spiral spring, and lifted by levers actuated by cams on the lay shaft.

According to their latest designs, Messrs. Crossley are building engines with one cylinder indicating up to 100 horse power, fitted with one or two flywheels, according to the requirements. Engines indicating 250 horse power are built with two cylinders placed opposite each other, their connecting rods working on the same crank pin. One specially heavy flywheel, supported by an extra bearing, is fitted to these engines for electric lighting purposes. Figs. 13 and 14 show the general external appearance of these engines. Small vertical engines, indicating up to 8 horse power, are built for various purposes. Those fitted with hoisting drums are geared to the latter by friction wheels, and an engine indicating 6 horse power may be calculated to lift 1,110 lb. at the rate of 60 ft. per minute, whereas a 2 horse power engine will lift 280 lb. at 80 ft. per minute. These figures depend to some extent upon the gearing used, and it is probable that under favourable circumstances both the weights and speeds might be increased, for it will be found from the above-quoted figures that the mechanical efficiency is from 33 per cent to 53 per cent, a somewhat low estimated efficiency for a direct-driving motor.

The ignition tube now used on the Crossley engines is a modified form of fig. 15. The central tube T is maintained at a bright red heat by means of a bunsen flame. A cylindrical cover R of non-conducting material serves to concentrate the heat. The action when working is as follows: During the compression stroke the small valve E closes upon the upper orifice by the upward movement of the lever L. The compressed mixture enters the cavity G. The moment ignition is required, the valve E descends by the pressure exerted by the spring S, when the lever L drops. The compressed gas then rushing into the tube T is ignited, and the flame strikes back into the cylinder. The valve E now

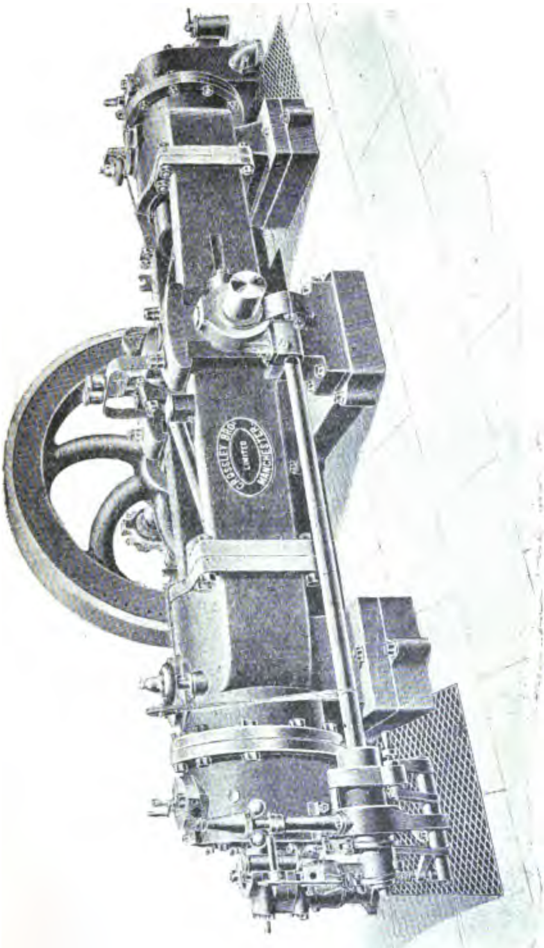


FIG. 14.—CROSSLEY'S DOUBLE-CYLINDER GAS ENGINE.

rises, and the burnt products in the tube T escape through the lower orifice of the valve E. In this design a rather long ignition tube is required, to ensure the fresh mixture reaching the hottest part of the tube T, by forcing to its upper closed end any of the burnt products remaining in the tube.

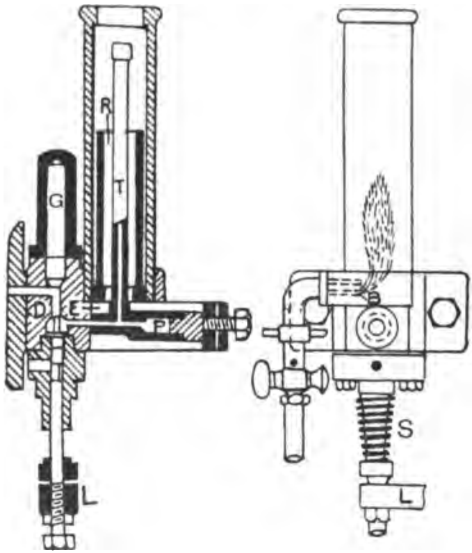


FIG. 15.—Otto Tube Igniter.

In 1893 Messrs. Crossley patented a self-starting arrangement for their gas engines, which consists essentially of a hand pump for forcing a charge into the cylinder when the crank is in a proper position for receiving an impulse. The exhaust cam on the lay shaft is so constructed as to prevent excessive compression when starting. When a charge has been pumped by hand into the cylinder, a timing valve is opened, and the charge fired by contact with the

ignition tube. When the engine has acquired a considerable momentum, the exhaust valve is made to close early, and so give the required compression for continuous running.

The latest improvement in the Crossley engines is said to be due to the introduction of a scavenging arrangement, by which the exhaust gases usually remaining in the clearance space are drawn away at the end of the stroke. The innovation is fully described in a paper read by Mr. J. Atkinson, M.I.Mech.E., before the Manchester Association of Engineers, from which the following description and illustrations have been derived.

Referring to fig. 16, which shows a diagrammatic view of the piston, cylinder, and path of the crank, we may take the

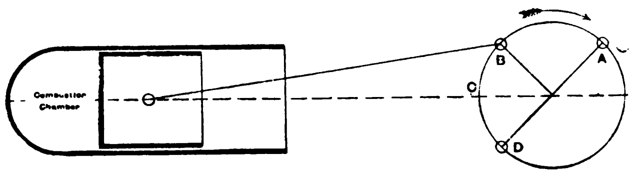


FIG. 16.—Diagram of Atkinson and Crossley's scavenging arrangement.

point A as the position of the crank pin when the exhaust valve opens for the discharge of the products of combustion. In the ordinary Otto engine this valve would close at the point C, and at the same time the air admission valve would open. But in the engines now made with Messrs. Atkinson and Crossley's scavenging arrangement, the exhaust valve remains open until the crank pin is at B, whilst the air admission valve opens at D. Thus, it will be noticed, both the air and exhaust valves are open during one-quarter of a revolution. When the exhaust valve first opens at A there is a pressure in the cylinder of about 35 lb. This is discharged into an abnormally long exhaust pipe, about 65 ft., through which it is driven by the returning piston. When the velocity of the piston approaches zero at the end of the instroke, this long column of exhaust gas, by virtue

of its inertia, causes a slight vacuum to be formed in the combustion chamber. At this time the air valve is opened (see position marked D), and a draught of fresh air is drawn through the combustion chamber into the exhaust pipe, sweeping out all the remaining burnt products and replacing them with fresh air. The exhaust valve closes at B, and the gas for the next charge may then enter through the gas valve.

The variation of pressure in the cylinder is clearly shown by the accompanying indicator diagram, fig. 17, taken with a $\frac{1}{8}$ in. spring, in order to show only the effect of the long exhaust pipe. The top line of the diagram has no significance, for the indicator piston is merely pressing upon a stop provided in order to prevent damage to the weak spring during the compression and explosion strokes. The

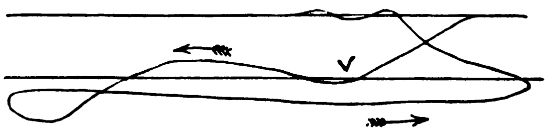
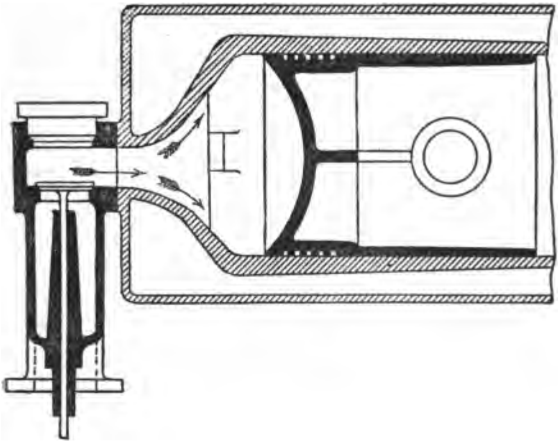


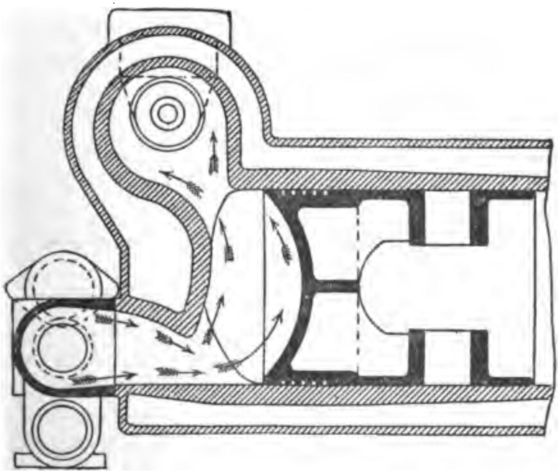
Fig. 17.—Diagram taken with $\frac{1}{8}$th spring.

pencil of the indicator leaves this horizontal line on the exhaust stroke of the engine, and the effect of the first puff of the exhaust is to set in motion a long column of gas, the inertia of which causes the slight vacuum shown at V. After this point the engine piston, travelling at its maximum velocity, causes a slight increase in the pressure, and the consequent acceleration of the gases in the exhaust pipe, which finally produces about $3\frac{1}{2}$ lb. vacuum, when the air valve admits fresh air. The suction stroke is completed at from 1 lb. to 2 lb. below the atmospheric line.

In order to assist the draught of air through the cylinder, the piston is specially shaped as shown in figs. 18 and 19. This drawing, together with fig. 20, also illustrates the usual style of mushroom valves used in nearly all modern gas engines. It will be noticed that no stuffing boxes are used,



Sectional elevation.



Sectional plan.

FIGS. 18 and 19.—Arrangement of air passages in Atkinson and Crossley's scavenging arrangement.

but that specially long, well-fitted spindles, working in long sleeves, are sufficient to prevent escapes. The valve spindles are always on the face subjected to the least pressure. Mr. Atkinson states that silencers and quietening chambers do not prevent the action of the exhaust, provided they are

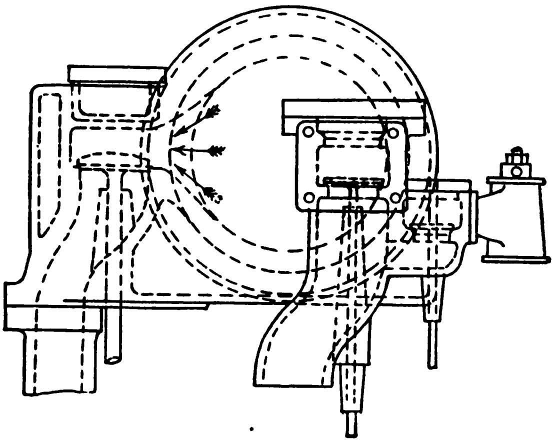


FIG. 20.—Mushroom valves used in gas engines.

placed at the end of a 65 ft. length of exhaust pipe of uniform diameter. No sudden enlargements are permissible within about 65 ft. of the engine, although drain pockets may be inserted for the collection of condensed moisture.

THE "STOCKPORT" GAS ENGINES.

The Stockport gas engine is manufactured by Messrs. J. E. H. Andrew and Co. Limited, Reddish, and works upon the Otto cycle. This firm commenced the manufacture of the "Bisschop" engine—a small motor not much used at the present—in 1878, and from that time have built over 7,000 motors of the Bisschop and Stockport design. The latest Stockport is a horizontal engine, the general features of which may be seen from the engraving (fig. 21).

The following table, supplied by the makers, is of use in the arrangement of details external to the engine itself :—

Effective horse power.	Revs. per minute.	Size of flywheels.			Standard size of pulley.		Overall dimensions, engine only.				Approximate weight.	
		Diam.		Width	Diam.	Width	Length.		Breadth.			
Vertical:		One: Ft.	In.	One: In.	In.	In.	Ft.	In.	Ft.	In.	Cwt.	Qr.
1½	220	3	1½	4	10	6	3	4	3	0	13	2
5	200	4	0	5	18	7	3	6	3	3	22	2
Horizontal:		Two:		Two:								
1	240	2	6	3½	9	5	4	9	2	6	9	2
2	220	2	11	2¾	10	6	5	3	3	0	13	2
3	220	3	2	3	14	6	5	9	3	9	18	0
5	220	3	6	4	16	7	6	9	4	0	27	3
7	220	3	11	4	18	7	7	0	4	3	32	2
9	200	4	3	5	21	8	8	0	4	6	40	0
11	200	4	3	5	21	8	8	3	5	0	45	0
13	190	4	8	5½	23	9	8	6	5	3	54	2
15	190	4	9	5½	23	10	8	9	5	6	58	2
17	185	5	2	6¼	27	12	8	10	5	9	66	0
19	185	5	3	6¼	27	12	9	0	6	0	70	0
22	185	5	3	6½	32	12	9	3	6	3	75	3
26	180	5	3	6	36	12	9	9	6	6	90	0
30	180	5	4	6	42	14	10	0	6	9	100	0
35	170	5	5	6	48	15	10	6	7	0	112	2
42	160	6	2	9	54	19	11	10	7	3	145	0
55	160	6	3	9	54	23	12	0	8	3	190	0
67	155	6	10	10	Special pulleys to suit.		13	0	8	6	210	0
80	150	7	0	10			13	3	8	9	245	0
100	150	7	2	10			17	0	10	0	360	0

The construction of the Stockport gas engine now built, may be followed by an examination of the accompanying

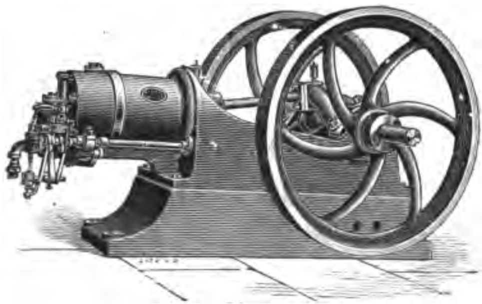


FIG. 21.—"Stockport" Gas Engine.

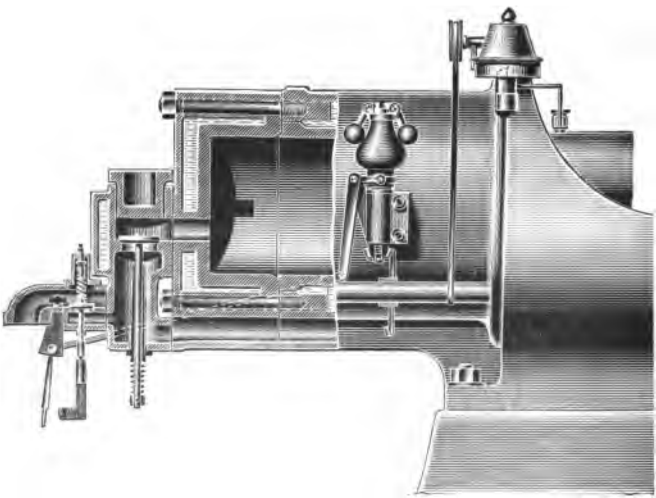


FIG. 22.—"Stockport" Gas Engine—Sectional Elevation.

drawings. Fig. 22 shows a section through the back of the cylinder, with the gas and air valves. It will be noticed

that the back of the cylinder is cast separately from the remainder of the cylinder and engine framing; and further (see fig. 25) that the metal forming the back of the water jacket is specially thin. By this arrangement the casting and boring of the cylinder is much facilitated, and in the event of frost getting to the jacket, when full of water,

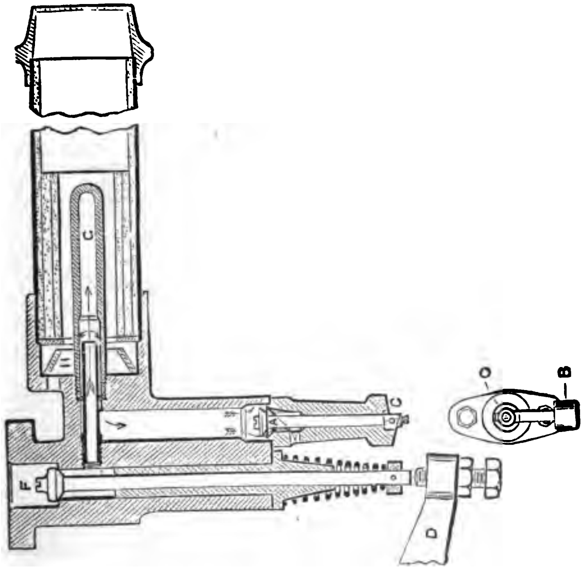


FIG. 23.—"Stockport" Gas Engine—Section of Timing Valve Bracket and Self-starter combined.

fracture due to the formation of ice will occur only at the thin wall of the jacket. This is easily and cheaply replaced, whereas if fracture occurred in the forward end of the cylinder, a large and expensive casting would be ruined.

In fig. 22 the supply of gas is coupled to the mouth of the pipe shown downwards. In this pipe is a wing throttle valve actuated by a lever from the governor. The gas

passes through the mushroom valve, when the latter is lifted by a cam, and passes into a chamber beneath the air valve. Here it mixes with the air, as both are drawn into the cylinder through the upper mushroom valve. This valve is lifted by a cam shown in fig. 24, and closes by its own weight, assisted by the spiral spring upon the spindle. When the mixture is compressed by the return stroke of the piston, it is forced up to the face of the timing valve shown in the end sectional view, also in detail in fig. 23. At the end of the stroke the timing valve F is pushed from

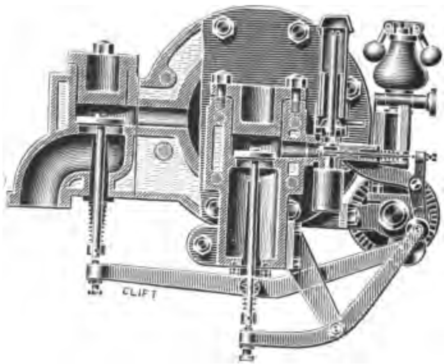


FIG. 24.—"Stockport" Gas Engine—Sectional End Elevation.

its seat by the lever D, and the gas entering the red-hot tube G is exploded. The timing valve may be adjusted by the set screw on the end of the lever D. If the explosion occurs too early, the set screw should be slightly withdrawn, if too late it should be slightly screwed forward and locked by the nut for that purpose.

The ignition tube G has an internal tube of smaller diameter entering its lower end. This affords an annular space communicating with the valve A. This valve, when down upon its seat, permits the escape of the gas in the tube after explosion, through a groove cut in the seating. This

renders the tube more certain in its action, for it is obvious that if the burnt products were pent up in the tube at a high pressure, the fresh mixture to be exploded would not enter, and would therefore not ignite. Messrs. Andrews make their ignition tubes of a special alloy of silver, which they maintain will resist the corrosive action of the explosive gases at high temperatures. The valve A, fig. 23, is of special use to assist the self starting of the engine. This is done in the following way. The engine is barred round until the connecting rod is on the top throw and about at right angles to the crank. The best position for starting is governed to a great extent, by the nature and quality of gas used, but

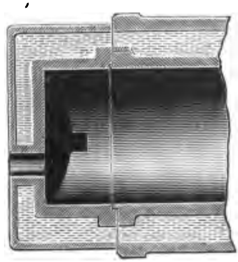


FIG. 25.—"Stockport" Gas Engine—Section of Cylinder End.

a few trials will enable the attendant to secure the best position for a given case. The exhaust valve lever is geared to the starting cam to prevent excessive compression at starting. After taking the precaution to admit gas gradually to the gas bags and to have the ignition tube at a bright red heat, the valve C, fig. 23, may be opened by raising the weighted lever B into a vertical position. The timing valve F will be open with the crank in the position for starting; consequently if gas is admitted to the cylinder by means of a special starter pipe, it will find its way, together with displaced air, through the valve F, up into the ignition tube, downwards as indicated by the arrows, and out at A.

This stream of air and gas continues until the mixture becomes rich enough in gas to explode. The engine will then start, and receive its next charge of gas through the main gas valve supplying the engine. The exhaust lever bowl is put to run upon the ordinary working cam; the valve A is closed by the action of the weighted lever B pressing the inclined plane at C. It will be noticed that if,

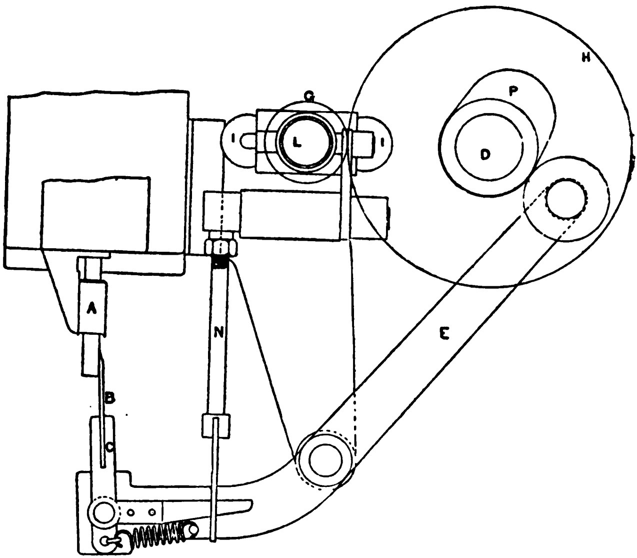


FIG. 26.—"Stockport" Gas Engine—Governing Arrangement.

in attempting to start, the gas enters the cylinder very rapidly, it may displace all the air first and not be sufficiently diffused to effect combustion when it passes into the hot tube. For this reason the gas is not turned fully on to the gas bags, so that it may have time to diffuse somewhat with the air in the cylinder before passing into the ignition tube. Immediately after the first impulse the gas should be fully turned on.

The engine is governed by first throttling the gas in the inlet pipe when the governor rises. Should the speed increase after throttling has taken place, the governor, still rising, brings the throttle-valve lever, fig. 22, in contact with the bell-crank lever actuating the gas-valve spindle, and so cuts off entirely the gas supply. The most recently

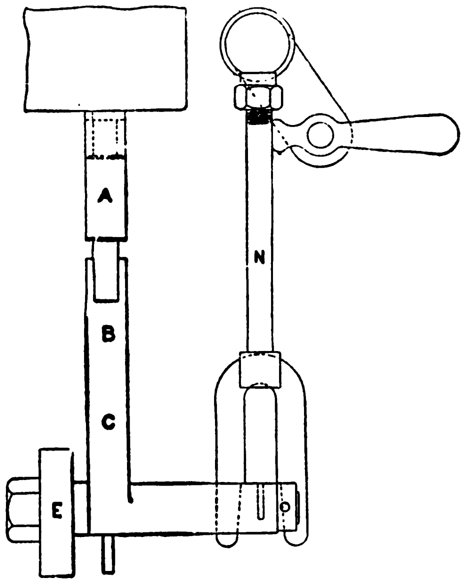


FIG. 27.—"Stockport" Gas Engine—Governing Arrangement.

designed governor used upon the Stockport engine is of the horizontal type, shown in figs. 26-27. A cam on the lay shaft drives the gas-valve lever E. At the end of E is a bell-crank lever, held in the position shown by the spiral spring. The rod N, actuated by the governor, terminates in a fork, which when moved sideways (see fig. 27) engages the end of the bell-crank lever, overcomes the tension of the

spiral spring, so that when the tripper blade rises it misses the notch under A on the gas-valve spindle, thus cutting out entirely the gas supply. The range of action of the throttle valve depends to some extent upon the width of the fork on the lever N, and when its limit is reached the gas is entirely cut off. The speed of the engine is raised

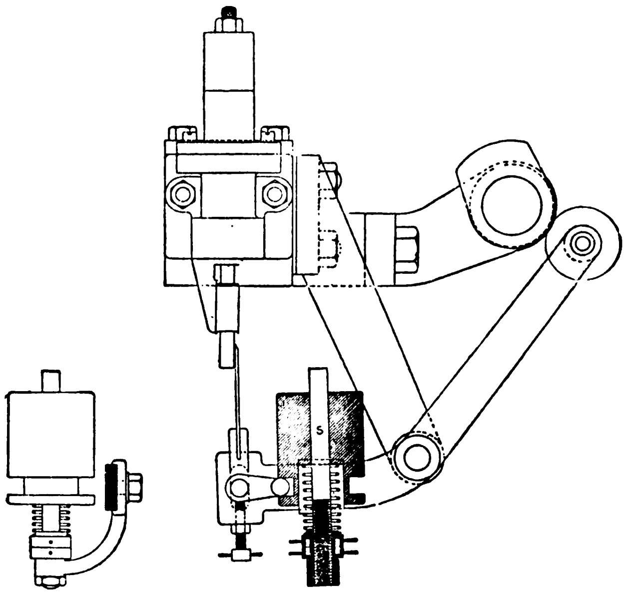


FIG. 28.—"Stockport" Gas Engine—Arrangement of Governor for Small Engines.

by compressing the spring on the governor spindle by means of two adjusting lock nuts.

The peculiar and sudden action of cam-driven levers renders it possible to entirely cut out the gas supply, when the speed increases, by a simpler mechanism than that just described. Fig. 28 shows a form of simple vibrating governor, which is fitted to the smaller engines of Stockport

design. The valve lever carries a vertical spindle S. Upon this spindle a cast-iron weight slides freely, but is supported by a spiral spring, the height of which may be adjusted by the two lock nuts shown. If the speed of the engine increases beyond the limit, the inertia of the weight upon

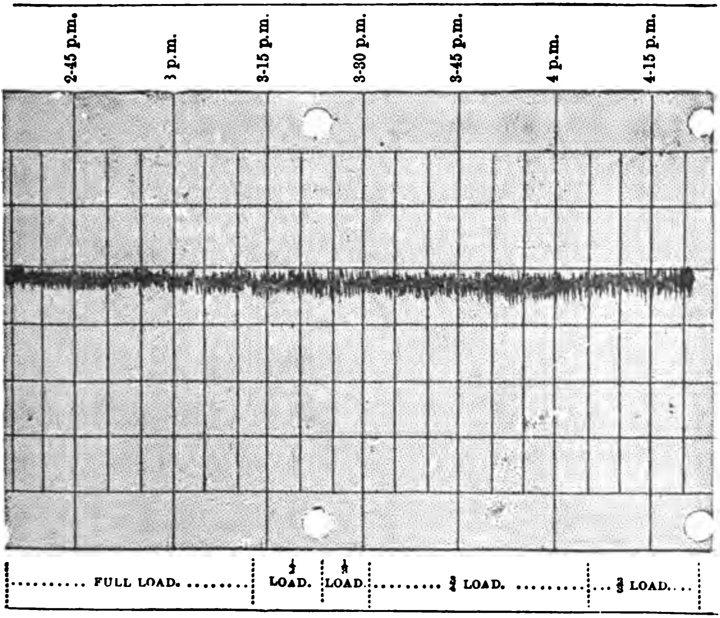


FIG. 29.—Moscrop Recorder Diagram.

the spindle S overcomes and compresses the spring supporting it. This in effect rotates the bell-crank lever slightly in a clock-wise direction, causing the tripper blade to miss the notch on the gas-valve spindle.

CHAPTER IV.

THE "GRIFFIN" ENGINE.

This engine has already been mentioned, and the cycle has been explained in the text relating to fig. 12. The Griffin engine is manufactured by Messrs. Dick, Kerr, and Co., of Kilmarnock, and since its introduction has been much simplified, and has had a considerable sale in England. Amongst other special features of the engine, the steadiness of running under varying loads is noteworthy. Fig. 29 is a Moscrop recorder diagram taken from a double-acting engine with single cylinder, developing 70 horse power on

COMPARATIVE CYCLES OF GAS ENGINES

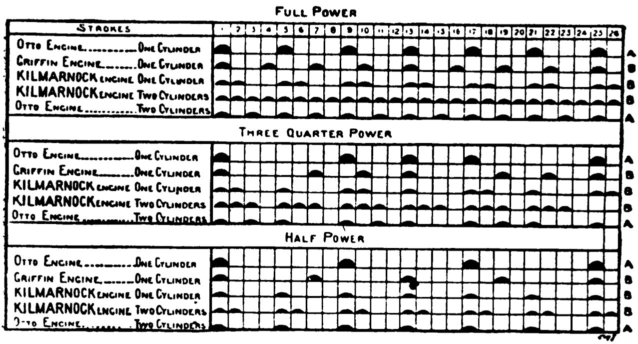


FIG. 30.

A. Single-acting engines. B. Double-acting engines.

the brake, and fitted with a special electric lighting governor. The distance between the horizontal lines represents a variation in speed of 5 per cent, and the distance between the vertical lines is traversed by the paper in five minutes. The trial lasted 1¾ hours, and the loads were varied between full load and one-third load, with a variation in speed of about 2½ per cent. The following diagram, fig. 30, supplied to the author by the makers.

shows the impulses given to the pistons of various types of engines running at full, three-quarter, and half power. It will be noticed here that the Kilmarnock engine, running at half load, has twice as many impulses as the ordinary Otto engine running at full load, and 3½ times as many at half load. All the engines constructed by Messrs. Dick, Kerr, and Co. larger than 12 brake horse power are double acting. The double-acting six-cycle engines, having two

FIG. 31.

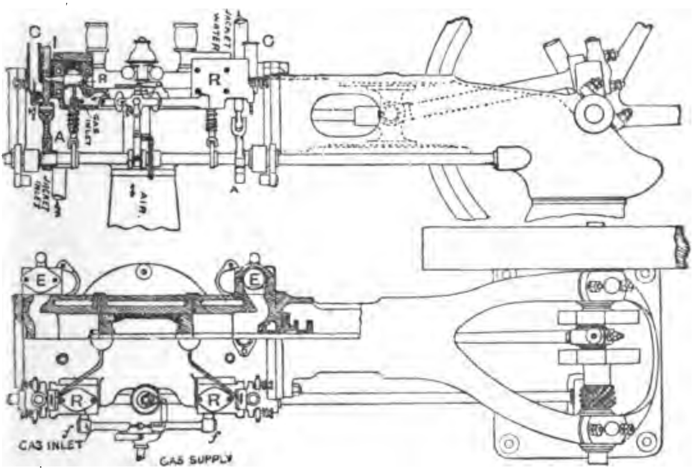


FIG. 32.

Sectional Elevation and Plan of Griffin Double-acting Gas Engine.

idle strokes, for scavenging purposes, are termed the "Griffin" motor, and the double-acting engines, dispensing with the idle strokes, are called the "Kilmarnock." In the latter type the turning effort exerted upon the crank pin is more regular, though, according to accepted theory, some loss in efficiency may be occasioned by the presence of burnt gases in the cylinder of this latter type.

The details of a double-acting engine will be understood on reference to fig. 31. This is an illustration of a recent design of Griffin engine. The engine is of the horizontal pattern, with balance weights on the crank webs. The side shaft is driven by a 3 to 1 worm gearing on the crank shaft, shown in fig. 32. The cylinder is closed at both ends, and is consequently kept free from the grinding action of dust, which in some factories may have a very serious effect upon an open-ended cylinder. The cylinder is water jacketed, together with the piston-rod gland. From the one side shaft all the valves and governor are driven. The two exhaust ports are shown at E E, and are operated by cams, acting on levers passing under the cylinder, lifting plain mushroom valves. On the opposite side of the cylinder are placed the governor gear, distribution valves, and igniting slides. At *f* in fig. 32 the exterior of the gas-valve boxes are shown. After passing through the valves at *f*, the gas meets air drawn up through the bed, as indicated by the arrows in fig. 31. Mixing with the air, it now passes through the inlet valve R into the cylinder, the valve R closes, and the charge is compressed in the combustion chamber and the space above the valve R.

The ignition slide I is now opened by the eccentric A, and makes communication between the flame in C and the compressed mixture. The governor controls the gas supply by throttling its passage through the valves in *f* and *f* until the gas is entirely cut off. Referring only to the back end of the cylinder, the sequence of operations is as follows: The gas valve *f* is opened by its cam, and gas, mixing with air, passes through R into the cylinder, R closes, and the charge is compressed on the return stroke of the piston. Ignition takes place, driving the piston forward. The exhaust opens, and the products of combustion are expelled. The exhaust valve E closes and R is opened, but as the gas valve *f* remains closed, air only is drawn into the cylinder. This in turn is expelled through the exhaust port E, and the cycle being completed, a fresh charge is again drawn

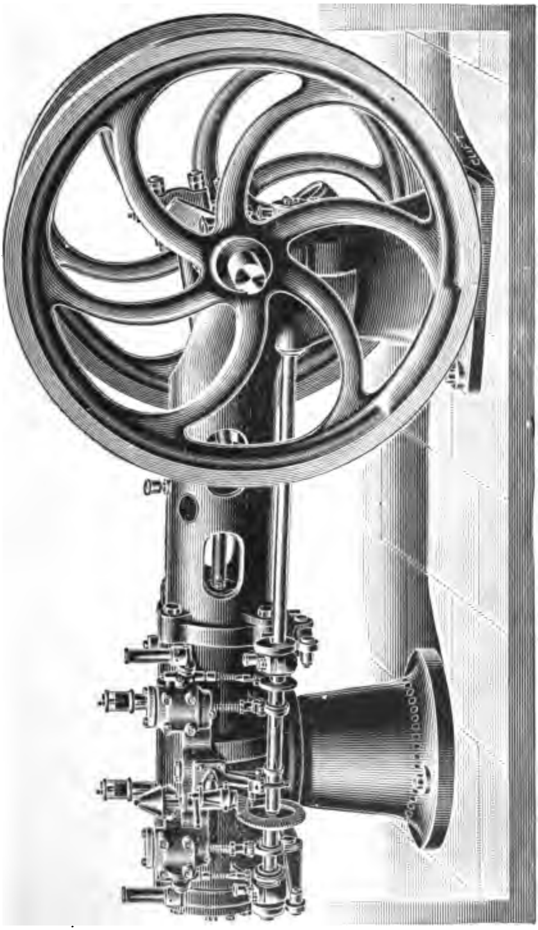


FIG. 33.—KILMARNOCK DOUBLE-ACTING GAS ENGINE

into the cylinder. Precisely the same action takes place in the front end of the cylinder, thus giving two impulses to the piston every six strokes.

Fig. 33 shows an outside view of the Kilmarnock engine, the details of the valves and igniting arrangements being somewhat revised. A timing valve, of the mush room pattern, is substituted for the slide, and a hot tube is substituted for the flame shown in the previous illustrations.

Fig. 34 is an illustration of a recent design of engines as supplied to the Corporation of Belfast for the generation of electricity for lighting purposes. Four of these engines, each indicating 120 horse power, are driving four 60 kilowatt dynamos, and two smaller engines, having single instead of tandem cylinders, are driving two 26 kilowatt dynamos. The efficiency of the engines and dynamos combined proved to be 76 per cent, and the gas consumption per electrical horse power was less than 24 cubic feet. This value might be somewhat reduced if rich coal gas were used, instead of, as in the present instance, the gas being composed of coal gas and enriched water gas. The tandem design has been adopted by Messrs. Dick, Kerr, and Co. because of its narrow width, and consequent economy of space. The increase in length, consequent upon the position of the tandem cylinder, is utilised by placing the cylinders between the dynamo and the engine crank shaft—an arrangement advocated upon a previous page. Each dynamo is driven by eight $\frac{7}{8}$ in. ropes, and for this reason it is essential to run the engine contra-clockwise, in order that the tight side of ropes shall be at the bottom. This precaution, as has been already pointed out, is not essential with a belt drive, but when ropes are used there is a danger of them leaving the grooves if not kept tight at the bottom. Each engine is fitted with a flywheel on the opposite end of the shaft occupied by the rope-driving pulley.

In this design of engine there is an explosion every stroke, but no scavenging strokes are made. In the front

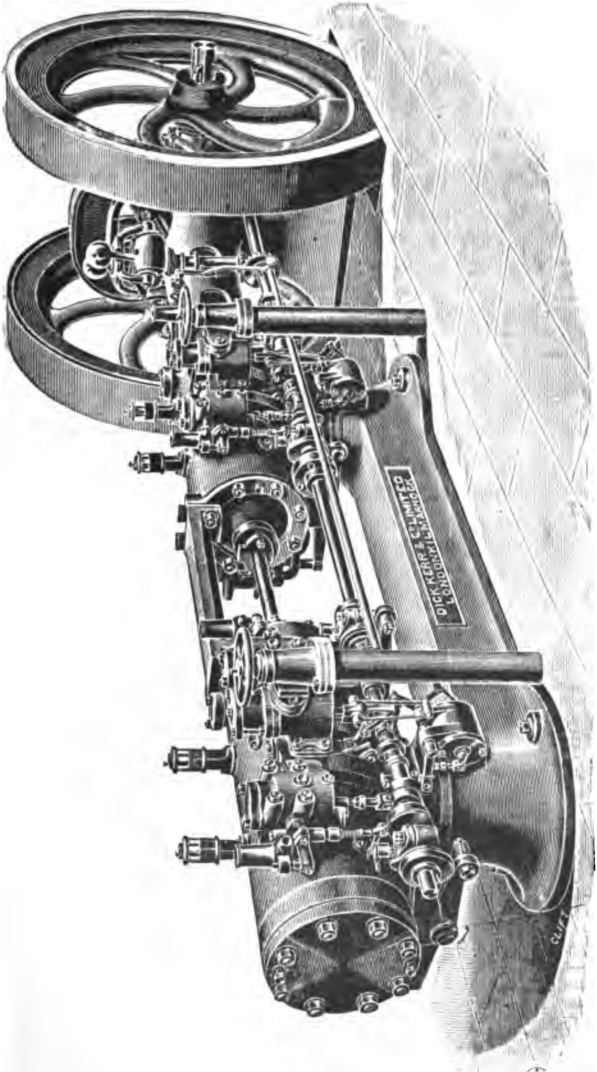


FIG. 34.—KILMARNOCK TANDEM GAS ENGINE.

and back of each cylinder a complete Otto cycle is carried out once in two revolutions in the following order: First, the charge in the back end of the back cylinder is fired; second, that in the front end of the forward cylinder; third, that in the back end of the forward cylinder; and lastly, the charge is fired in the front end of the back cylinder. The governing is carried out by a special electric light governor, reducing the quantity of gas supplied to each of the cylinders. This method is found preferable to one in which the gas supply is entirely cut out. At Belfast the starting of the engine is effected by utilising electric energy stored in the batteries, by sending a current through the dynamos, converting them for the time being into motors, and so driving the gas engines, until a charge is drawn into the cylinder and ignited.

The exhaust pipe of each engine communicates with a main exhaust pipe, to which a silencer is fitted, but it is found that there is less probability of noise when two or more engines are working than when only one is exhausting, for the flow of exhaust gas is then more regular, and does not cause that coughing noise often so objectionable.

Besides a large circulating tank for supplying the water jackets, provision is made for admitting water from the town mains direct to the jacket; and as a still further precaution, each engine is provided with a circulating pump for driving water through the jacket.

At Coatbridge,* near Glasgow, Messrs. Dick, Kerr, and Co. have supplied engines for driving the electric light, worked by producer gas. In this installation each engine has two cylinders placed side by side instead of tandem. The starting of the engines is effected by utilising the pressure in the steam boiler used in connection with the gas-producing plant, about which more will be written. A special valve is fitted to the back of one cylinder of the engine, and is operated

* Since the publication of the first edition of this book, these engines have been removed and steam substituted.

entirely by a hand lever. A steam pipe is connected to each of these valves, and steam enters the cylinder when the operator opens the starting valve. The operator, performing the work of the eccentric on a steam engine, opens and closes the starting valve until the engine has acquired a momentum sufficiently great to draw into the cylinder, and compress, a charge. Ignition then takes place. The largest gas engine made by this firm indicates over 700 horse power, and is specially interesting, inasmuch as it is constructed upon a compound principle. The engine somewhat resembles a large triple-expansion horizontal steam engine, and has two 10 ft. diameter by 14 in. face flywheels. Two 21 in. diameter by 30 in. stroke cylinders are placed one on each side a 32 in. diameter cylinder having a 36 in. stroke. One of the high-pressure cylinders only exhausts into the central cylinder, the other being kept separate for starting the engine by the application of steam pressure, as above described.

CHAPTER V.

THE TANGYE ENGINE

MESSRS. TANGYE'S gas engines work upon the Otto cycle, but improvements have been made in the igniting and governing gear, and in a self-starting arrangement. Fig. 35 shows an outside view of an engine which will give a maximum indicated horse power of 115. Messrs. Tangye construct single-cylinder engines, indicating ½ H.P. to 150 H.P., all of which work with either town or other gases made by their own producer plant. The combustion chamber is specially formed to prevent shock to the working parts during explosion, and also assists in making the charge more homogeneous. Messrs. Tangye claim that these improvements render their engines specially useful for electric lighting, a large number having already been made for that purpose. It is stated that, working with gas

costing 2s. 6d. per 1,000 cubic feet, a 30 H.P. engine will produce 1 kilowatt for each penny expended, which works out to about 2 cubic feet of gas per 10 candle power lamp. The weight of anthracite coal used in the producers for running an engine of about 70 brake horse power is said to be less than 1 lb. per B.H.P. per hour.

The large engines are now started with a new and very powerful starter, which is capable of starting the engine against half its working load. A charge of gas and air is pumped by hand into a detached receiver, there being an excess of gas present, thus rendering the charge inexplosive in the receiver. The engine is barred round into the position for the commencement of a forward stroke. The charge in the receiver is pumped up to a pressure of about 40 lb. per square inch. At this pressure communication is made with the working cylinder, and the charge enters the combustion chamber, and by mixing with the air present in the combustion chamber forms a strong explosive mixture. This, however, is at first prevented from igniting by the closed ignition valve being tightly held upon its seat by a lever operated by a cam upon the side shaft. Under the initial pressure of the gas entering the combustion chamber the piston moves forward; the cam then releases the ignition valve, and explosion takes place.

By setting the engine in motion before igniting the charge, excessive stress upon the working parts is greatly avoided. Fig. 36 is a drawing of the self-starter patented by Mr. C. W. Pinkney, to whom many other improvements in Messrs. Tangye's gas engines are due. In the drawing, the ignition tube I and starting pump are shown together; this arrangement is, of course, subjected to modifications necessary in special cases. The valves *a* and *b* are the suction and delivery valves respectively of the starting pump P. The valve *a*, shown separately, is contained in the same valve box as *b*, and is in communication with the passage *c*. A gas pipe is coupled to *g*, and by the small port shown the gas mixes with the air drawn in

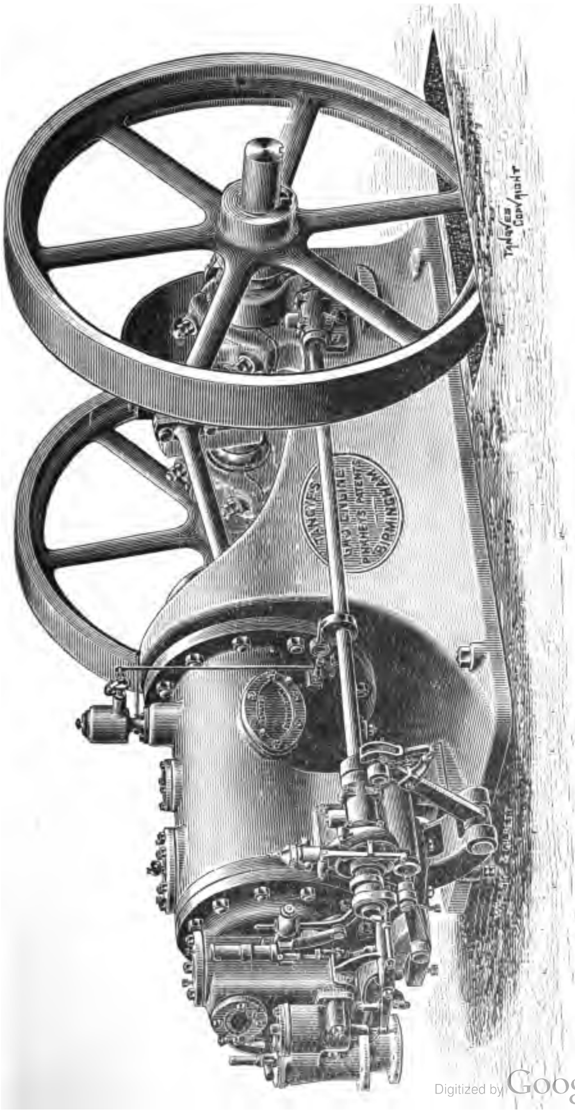


FIG. 35.—THE TANGYE GAS ENGINE.

through the valve *a*. The gas port terminates in an annular space, to facilitate the mixing with the air. The proportion of these valves is such as to admit a mixture of about four volumes of air to one of gas. This may be delivered to the receiver, as previously mentioned, or may be pumped directly into the combustion chamber. In the latter case

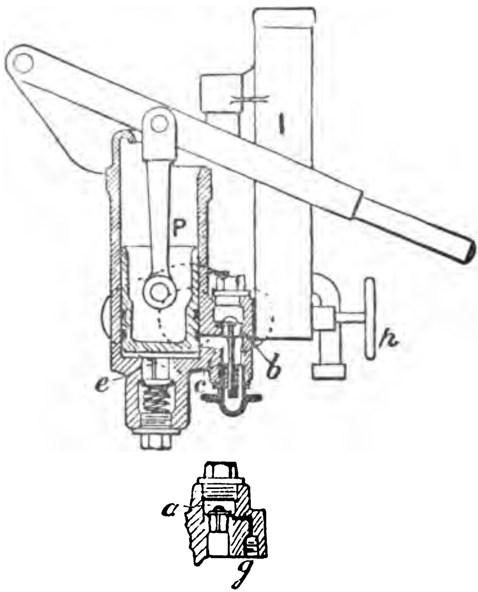


Fig. 36.—Pinkney's Self-starter for Gas Engines.

the handwheel *h* controls admission to the ignition tube, and the engine is arranged to fire only when the hand wheel is turned. The valve *e* is held up to its seating by a spiral spring beneath it, and the spring is adjusted so that the valve *e* may open when the desired pressure in the pump and combustion chamber, or receiver, has been

acquired, thus giving an indication that the mixture is ready for firing. In connection with this starter, Mr. Pinkney has patented an arrangement for preventing the movement of the engine piston when the pressure of the hand pump is first felt in the combustion chamber ; for it is obvious that, when the hand pump delivers directly into the combustion chamber of a small engine, the crank will slowly move towards the forward dead centre, and be in that position when the charge is ready for firing. To obviate this, a stud upon the crank web engages with a spring catch of sufficient strength to hold the crank in its starting position until the greater pressure of the explosion causes its release. The bar, with its catch, then falls clear of the crank as it revolves. This gear is only suitable for

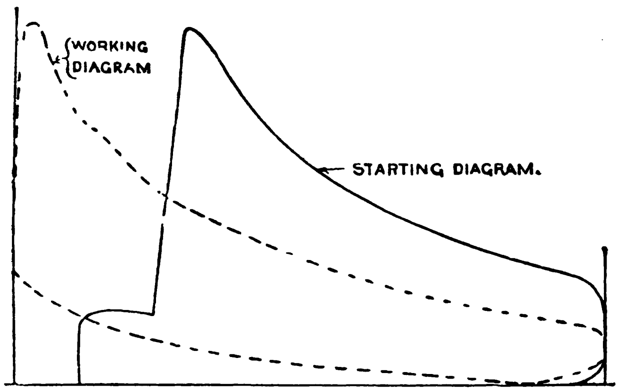


FIG. 37.—Starting and Working Diagram taken from Tangyes' Gas Engine.

small engines fitted with self-starters, larger engines being fitted with a special receiver to contain the charge until it is admitted to the combustion chamber by means of a valve operated by hand.

An indicator diagram taken from an engine of this latter type is shown in fig. 37. It is seen from this diagram that

the engine piston is standing in a position equal to one-ninth of its forward stroke before communication is made between the combustion chamber and the receiver. The pressure immediately rises when the communicating valve is opened to about 40 lb. per square inch. The diagram also shows that two-ninths of the forward stroke are completed before ignition takes place. The remainder of the full-line diagram shows the explosion and expansion during the first revolution. The dotted diagram has been taken after the engine has acquired its proper working speed. The latter shows a maximum pressure of 200 lb. per square inch, and a compression pressure of 72 lb. per square inch.

Messrs. Tangye's engines have a specially hard metal liner fitted to the cylinder, which can quickly be replaced. Lubrication is effected by motion derived from the side shaft. A ratchet wheel, shown on the lubricator in the external view of these engines, slowly revolves, and thereby raises a plunger against the resistance of a strong spiral spring. Oil follows into the plunger chamber, and is injected into the cylinder by the release of the plunger from the top of its stroke. The worm wheels driving the side shaft are cut from solid metal, and run in an oil bath, the casing of which can easily be removed for examination of the wheels. The governor is of the centrifugal high-speed type, and acts directly upon the gas supply without throttling.

THE FIELDING GAS ENGINE.

This engine is made by Messrs. Fielding and Platt, of the Atlas Works, Gloucester, and is constructed upon the Otto cycle. Fig. 38 shows an outside view of the engine. The valves, which are of the plain mitre type, are driven in the usual way from the crank shaft, by means of cams on a side shaft. The governor is of the ordinary high-speed type, and is driven from the side shaft by bevel wheels. Another type of governor used on Messrs. Fielding and Platt's earlier

FIG. 38.—THE FIELDING GAS ENGINE.

engines was actuated by a cam on the side shaft. In this case the governor consisted of a simple dashpot and piston, the rod of which was connected to a lever working the gas

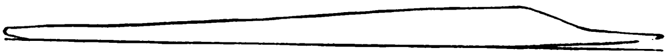


Fig. 39.—Indicator Diagram from Fielding and Platt's Engine, running with no load.

valve. If the speed of the engine increased beyond that to which the dashpot was adjusted, then the cam left the piston at the top of its stroke, coming round into its lifting

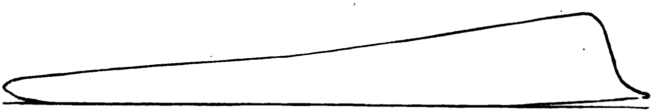


Fig. 40.—Indicator Diagram from Fielding and Platt's Engine, running at about one-third load.

position before the piston had time to descend ; consequently the gas valve was not opened.

A feature in the governing of this firm's electric lighting

Fig. 41.—Indicator Diagram from Fielding and Platt's Engine, running at nearly full load.

engine is illustrated by the accompanying indicator diagrams, figs. 39, 40, and 41. The governor is of the ball type, and regulates the gas and air supply, so that even when

running light an explosion is never missed. This arrangement, though possibly slightly increasing the consumption of gas, undoubtedly conduces to steady running under a fluctuating load. It is stated by Messrs. Fielding and Platt that the variation in speed between light and full load is about 3½ per cent. The gas consumption in an engine of this type, of 100 B.H.P., is claimed to be under 20 cubic feet per hour per B.H.P.

Messrs. Fielding have recently patented an efficient system of starting their larger gas engines by the use of compressed air. When the engine is about to be stopped after its first run the gas is turned off, and the piston and engine cylinder, by a slight alteration in valves, is converted into an air compressor, which discharges into a reservoir until a pressure of about 60 lb. is obtained. In this way the energy stored in the flywheels is utilised until the engine comes to rest. The charge of air thus obtained is sufficient to start from six to twelve times. The starting is effected in the following way: The crank is barred just over the back dead centre, and gas is admitted by a hand gas tap to the combustion chamber. The displaced air is allowed to escape through an open cock until gas which will burn with a clear blue flame begins to issue from the open cock. These cocks are now closed, and compressed air from the reservoir is admitted to the combustion chamber, forming a rich mixture. The igniting valve is now opened, and a powerful impulse is given to the piston. It is said that the engine will start by this means even against half the full load.

THE "PREMIER" GAS ENGINE.

The "Premier" gas engine, patented by Mr. J. H. Hamilton, B.Sc., is manufactured by Messrs. Wells Brothers, Sandiacre, near Nottingham. An external view is shown in fig. 42, with modified valve gear. Its characteristic feature is the scavenging arrangement. Fig. 43 is a sectional view of the engine, and from it the working of the scavenging stroke

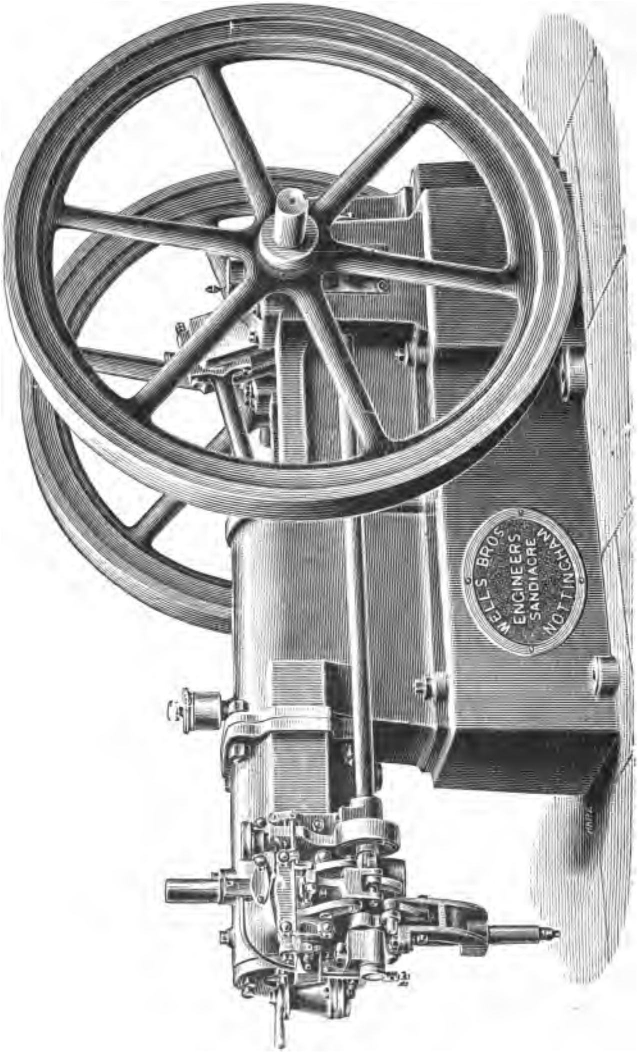


FIG. 42.—WELLS' PREMIER GAS ENGINE.

may easily be followed. There are two pistons, C and D, cast together with a cylindrical connection. The piston C receives the impulse due to the explosion of the charge, whilst the piston D acts merely as an air pump. The exhaust valve E is actuated by a lever driven by a cam on the side shaft (see fig. 44). The gas and air valves are driven by one lever, and are arranged, as shown, upon the same spindle. The gas valve is made an easy fit upon the air-valve spindle, but is held up to its seat by a spiral spring. Upon the outstroke of the pistons, air is drawn through rubber or leather valves in the bed plate, shown at A. The chamber marked A has a free communication with the chamber marked B. When the air valve is depressed beyond the position shown in the drawing, the gas valve is opened by contact with the collar shown upon the spindle. The exhaust valve E being closed during the outstroke of the pistons, air, mixing with the gas, enters the combustion chamber, as indicated by the arrows. When sufficient gas to form a charge has entered, the air-valve spindle slightly lifts, closes the gas supply, but still admits air to complete the charge. Upon the instroke of the pistons, the charge is compressed in the combustion chamber to about 60 lb. per square inch; at the same time the air entrapped between the two pistons becomes compressed to about 9 lb. per square inch. Thus the air valve has upon its lower face a pressure of 60 lb. per square inch, and upon its upper face 9 lb. per square inch; the difference, 51 lb. per square inch, tends to keep the valve tight upon its seat. In order to control the action of the air valve, and to make its movement independent of the difference of pressure upon its upper and lower faces, a strong spiral spring is attached to the lever L, fig. 44, tending always to force the air valve on its seat. The cam driving L gives the air valve a long and a short stroke alternately, the former for the admission of the charge, the latter for the admission of air only.

When the charge is compressed it is fired by a timing valve, and both pistons travel outwards. During this out-

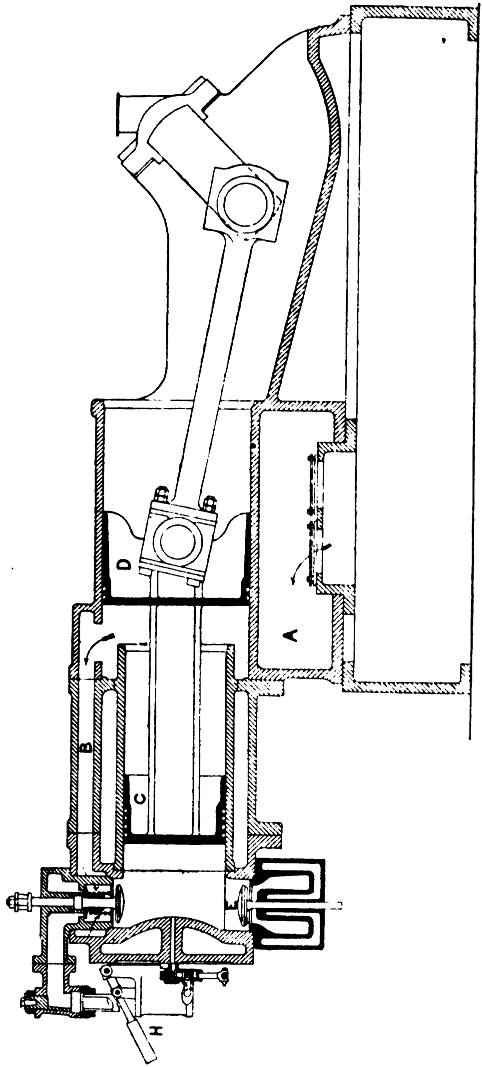


FIG. 43.—LONGITUDINAL SECTION OF PREMIER GAS ENGINE.

stroke the air already compressed by the larger piston D expands, and gives back to the engine the energy stored in it by compression. At the end of the outstroke the exhaust valve opens, and releases the pressure in the combustion chamber. It is evident that so long as the pressure in the combustion chamber during the exhaust stroke is nearly atmospheric, the piston D will displace a volume of

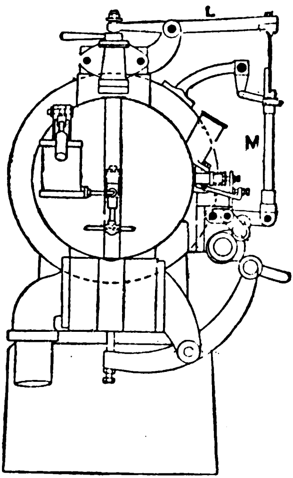


FIG. 44.– End View of Premier Gas Engine.

air and drive it through the combustion chamber into the exhaust pipe. This effectually clears the combustion chamber of its burnt products, and this done, the exhaust closes, and the cycle is repeated. The indicator diagram shown in fig. 45 has been taken from the working cylinder, whilst the diagram shown in fig. 46 has been taken from the pumping cylinder. Following the direction of the arrows on fig. 46, it will be seen that little or no work is lost in compressing the entrapped air in the chamber B, for

the air-expansion curve nearly coincides with the compression curve. At the commencement of the exhaust stroke the pumping piston raises the air pressure, the air valve opens, and air is forced into the combustion chamber *against the*

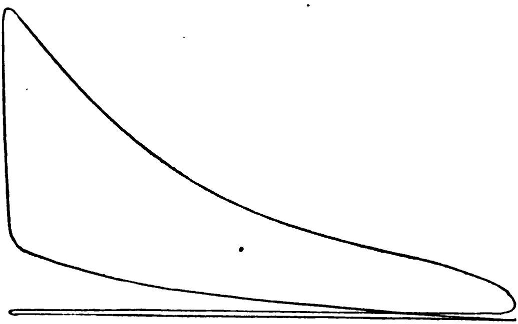


FIG. 45.—Working Diagram from Premier Gas Engine.

pressure of the exhaust. This point is shown on the diagram at P; the area P A S E of this diagram gives the work done by the pump in retarding the motion of the engine; and the net indicated horse power is found by subtracting from the

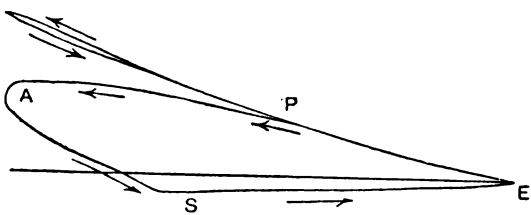


FIG. 46.—Pumping Diagram from Premier Gas Engine.

I.H.P. of the working cylinder the I.H.P. of the pumping cylinder. It must be remembered that the area P A S E of the pumping diagram occurs only at every alternate revolution; consequently, in obtaining the H.P. the revolutions

per minute must be divided by 2. It must also be noted that the effective area of the piston D, fig. 43, is not its whole area, but only that which remains after subtracting from it the area of the piston C.

In engines of this type it is extremely important, from the makers' point of view, to have a free exhaust pipe, for cases are not unknown in which the undue throttling of the exhaust pipes has so raised the back pressure as to cause the burnt products to enter the chamber B as soon as the air valve opens, thus entirely preventing the scavenging action. It is, for another reason, highly important to keep the exhaust pressure low, for the area of the pumping diagram depends upon it, and represents lost work.

This engine is started by means of a hand pump in the following way: The bowl on the exhaust lever is made to engage with a projection on its cam, so that the compression of the charge may be relieved. The engine is then barred round until the crank has turned through about 45 deg. of its forward *working* stroke. It is very important that the engine should start from such a position that all the valves are ready for the impulse, for it is obvious that by careless handling the engine might be started from the position corresponding to the suction stroke. In the latter case the exhaust valve would not open at the end of the stroke, and the burnt products would enter the chamber B, and probably damage the valves at X. When the engine is in the correct position, the timing valve is closed by a short lever inserted by hand. Gas is then turned to the hand-pump suction valve, and two or three strokes taken with the lever H. The gas supply to the pump is then turned oft, and the main gas cock to the engine opened. The hand-pump lever is again worked, and the pump delivers air to the combustion chamber. As soon as the engine piston commences to move forward under the pressure of the charge, the timing valve is released by hand, the charge enters the ignition tube, and a strong forward impulse is given. When the engine has acquired some momentum,

the exhaust bowl is put into its working position, and gives the correct compression to the charges. The relative number of strokes of the hand pump when drawing gas and air respectively must be determined for each case by trial, as the quality of gas varies considerably.

The governing of the engine is effected by the movement of the rod M, fig. 44, actuated by an ordinary high-speed centrifugal governor. When M is moved to the right its knife edge acts upon the gas lever L at a greater distance from its fulcrum; consequently the gas valve is not fully

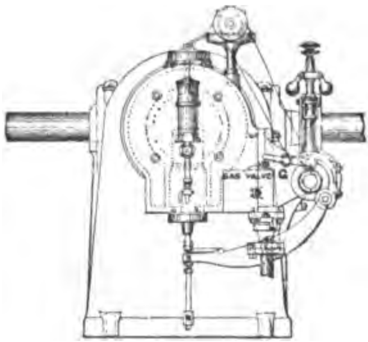


FIG. 47.—End View of "Forward" Otto Gas Engine.

opened. It is merely a matter of arrangement whether the gas supply be entirely cut off or gradually decreased. In fig. 44, the lever M would first throttle the gas supply, and after a further increase in speed cut it off entirely.

The ignition tubes supplied by this firm are made in nickel, and will stand continuous use for several months.

The "Forward" gas engine is manufactured by Messrs. T. B. Barker and Co., of Birmingham. Fig. 47 shows a sectional elevation and end view of this engine. The cylinder is 10 in. diameter by 20 in. stroke, and running at a speed of 180 revolutions per minute the engine will develop about 25 I.H.P., and about 20 on the brake.

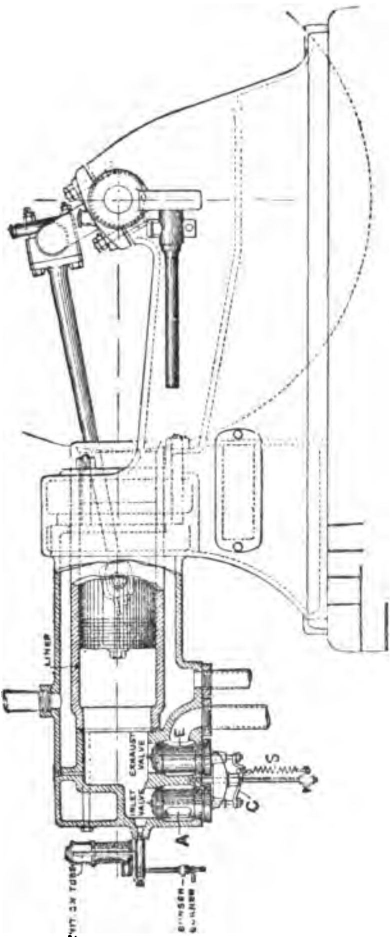


FIG. 48.—LONGITUDINAL SECTION OF "FORWARD" OTTO GAS ENGINE.

An important feature in this design is the reduction of port surface to a minimum. It will be noticed that the air and exhaust valves A and E, fig. 48, are placed as near the combustion chamber as possible. They are fitted to separate seatings, as shown at A and E. The gas is admitted by the valve G, fig. 48, and mingles with the air on its way to the valve A. The spring S is connected to a bar, one end of which rests upon the crossbar C, and the other upon the levers driving the valves. By this arrangement one spring suffices. Messrs. Barker and Co. do not use an ignition valve, but rely entirely upon the compression of the charge into the ignition tube for the correct timing of the explosion. The cylinder is formed by a liner of special cylinder metal, having an asbestos joint where it is attached to the combustion chamber. This to some extent prevents the conduction of heat from the combustion chamber to the fore end of the cylinder. The joint at the front end of the cylinder is made with indiarubber, and the liner is free to expand in the direction of its length as it becomes heated. As the function of this joint is merely to retain the jacket water, there is no necessity for an exceedingly tight fit. The crank shaft is of forged Siemens-Martin steel, and is fitted with phosphor-bronze bearings, the mean pressure upon the bearings being 100 lb. per square inch during the working stroke. The crank-pin pressure allowed is about 370 lb. per square inch, whilst the piston pin carries a mean pressure of 530 lb. per square inch.

These figures are worked out on the mean effective pressure given by the indicator diagram during a working stroke. The piston is packed with three cast-iron rings ⅝ in. wide, having a baffle groove ⅛ in. wide cut in each ring, dividing the bearing surfaces into two bands, each ¼ in. wide. The pressure exerted upon the walls of the cylinder amounts to about 5 lb. per square inch of surface. The piston speed of this engine running at 180 revolutions is nearly 600 ft. per minute, and the velocity of the inlet and outlet through the valves is about 100 ft. per second.

The following figures are quoted from a trial of a "Forward" engine of 30 B.H.P., conducted by Messrs. Morrison and Lanchester for the City of Birmingham Gas Dept. in 1894. When running with no load, the ratio of air to gas by volume was 14·5 to 1; when running with full load, the ratio was 9·9 air to 1 volume of gas. In the latter case the products of combustion were not swept out as the engine fired every cycle; consequently the volume of air plus products to that of the gas was 11 to 1. The gas consumption in cubic feet per hour per I.H.P. was 17·75, and per B.H.P. 21·05 cubic feet. To this should be added a total of 5·25 cubic feet per hour used by the burner for heating the ignition tube. The mechanical efficiency worked out to 84 per cent, and the thermal efficiency calculated on the I.H.P. was 22·2 per cent, and 18·6 when calculated on the B.H.P. The engine was governed when running light by cutting out the gas supply.

THE "GARDNER" GAS ENGINE.

The "Gardner" gas engine is made by Messrs. L. Gardner and Sons, of Barton Hall Engine Works, Patricroft. The engine differs from most gas engines in that the gas, air and exhaust valves are actuated by means of eccentrics instead of by cam driven levers, worked from a side shaft. The engine works upon the Beau de Rochas cycle. The eccentrics E, fig. 49, are driven from the crank shaft by means of spur and pinion gear, the spur wheel being twice the diameter of the pinion on the shaft. The air and gas valves are placed side by side (see fig. 50) and are operated by means of the same eccentric through a mechanism which admits of the gas being cut out when the speed is raised beyond the normal. In order to effect this the stud S, fig. 51, carries a rocking link L driven by the eccentric, the upper end of which is forked. That portion of the forked end which operates the air valve (see A, fig. 49) is carried upwards in line with the lower portion of the link to the air valve spindle, and when moved forward by the eccentric, opens the air valve. That portion of the fork operating the gas valve is bent away from

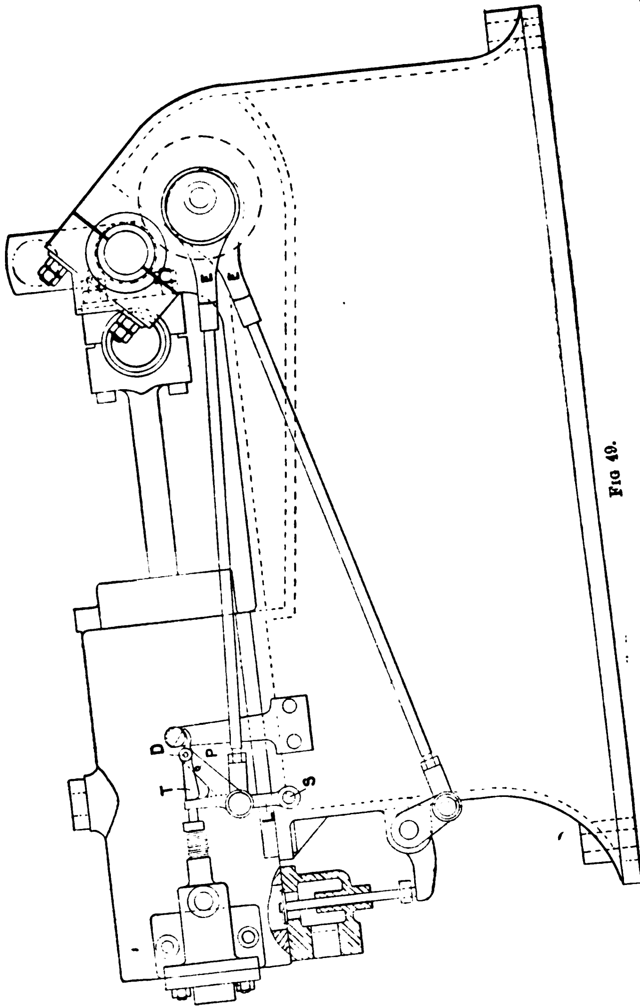


FIG 49.

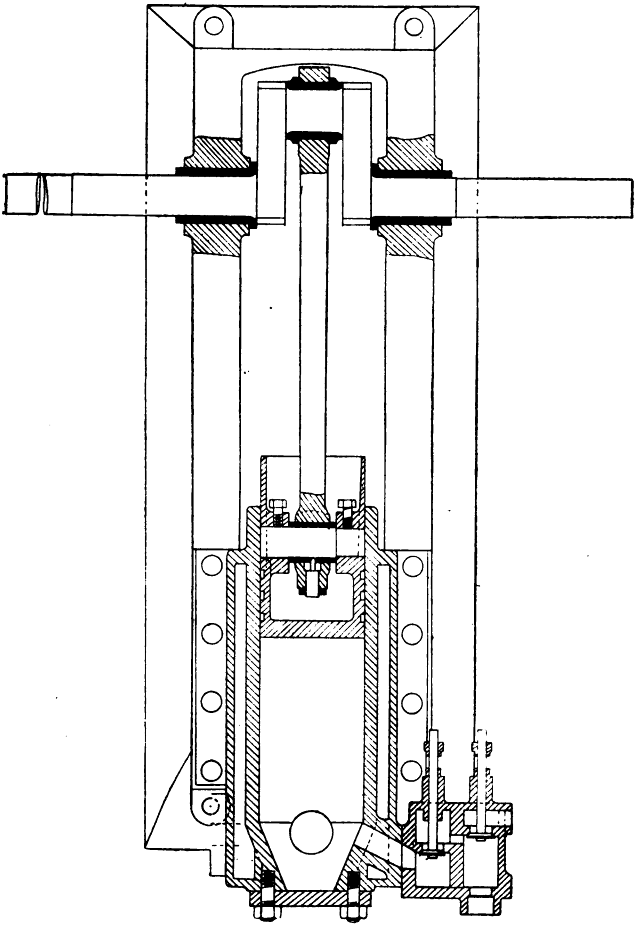


FIG. 50.

the valve spindle and terminates with a stud D carrying the trigger T. This trigger rests upon the stop pin P when drawn away from the gas valve spindle, but on being pushed forward, the trigger heel engages with a kicking stud K, which brings the trigger into a horizontal position, thus causing the trigger blade to engage with the gas valve spindle and open it the required amount. If the speed of the engine should increase beyond that for which the mechanism is set, the blade would be kicked above the gas valve spindle and so fail to actuate it. The speed at which this takes place may be regulated by altering the tension of

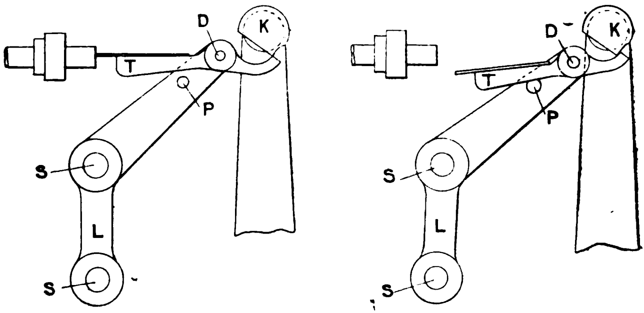


Fig. 51.

the torsion spring acting on the trigger. A readjustment of speed may be effected while the engine is running. The action of these parts will be understood from the drawings.

The engines are made in sizes ranging from ½ H.P. to 20 H.P.

The following figures have been sent to us by Messrs. Gardner and Sons :—

No. 1 Engine—1¼ Maximum Brake Horse Power.

Cylinder diameter	3½ inches.
Piston Stroke	5 inches.
Revolutions	350 per minute.
Average Piston Speed	291·5 ft. per minute.
Gas Consumption (including ignition burner)	30 cubic feet per B.H.P. hour.
Gas Consumption of larger engines (including ignition burners)	19 cubic feet per B.H.P. hour.

In an engine of the size above-mentioned the ignition burner necessarily consumes a considerable proportion of the whole volume of gas.

THE "WESTINGHOUSE" GAS ENGINE.

The British Westinghouse Electric and Manufacturing Company Limited has recently introduced a gas engine, which has some special features in design rendering it an effective and economical engine. We have formerly advocated the advantage of regulating not only the supply of gas by means of the governor gear, but also the simultaneous regulation of the air. This method of regulation has been aimed at by other makers, but we think that the mechanical details of the "Westinghouse" engine give effect to this method of governing in a way which has not yet been equalled or excelled.

By means of the series of photographs, kindly placed at our disposal by the makers, the general appearance of the engines will be understood. It will be seen that the design differs from that usually adopted in gas engines, in that the cylinders are vertical and follow the pattern of the "Westinghouse" steam engine rather than the usual form of gas engine. The Beau de Rochas cycle is used in each cylinder, but as the larger engines are built with two and sometimes three cylinders, impulses are given every revolution or every two-thirds of a revolution as the case may be. Referring to the details depicted in fig. 52, the cam shaft A is driven from the main shaft by the usual two to one gearing, and a cam upon it operates the lever G, which acts upon the stem of the exhaust valve E. Motion is transmitted from the shaft A to the shaft B, which latter also carries a cam for opening the inlet valve J by means of the lever C. The relative proportion of air to gas may be fixed by the attendant in a manner about to be described, but when once this is determined the engine continues to govern itself by increasing or diminishing the *total volume* of the mixture admitted to the cylinder, the *quality* being always the same.

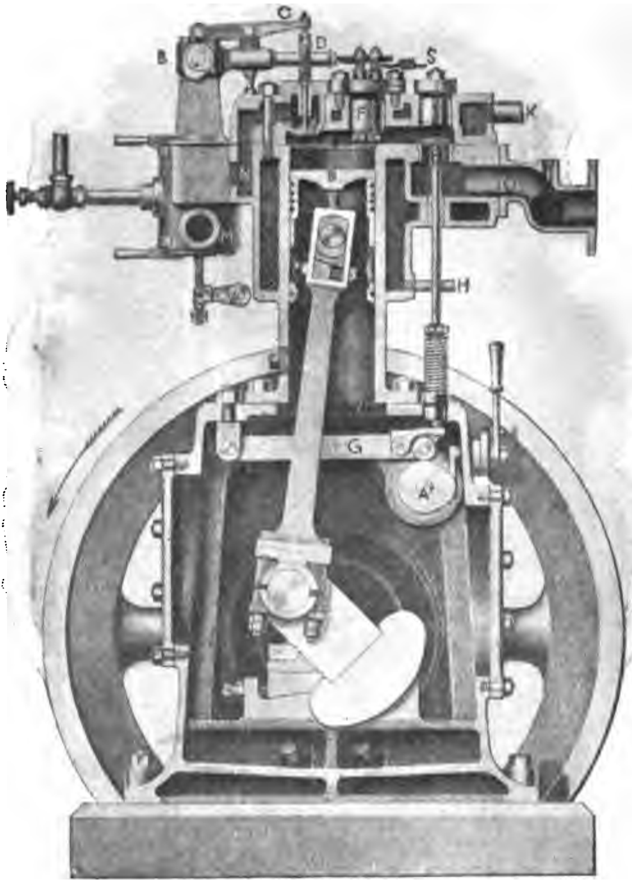


FIG. 52.

The ignitions of the charges are effected electrically by means of the sparker F. This has two pairs of sparking points, one pair being reserved for emergency. Should the sparker in action suddenly fail, the other pair may be at once switched in. The time at which the spark crosses is determined by means of the plunger piece D being moved forward

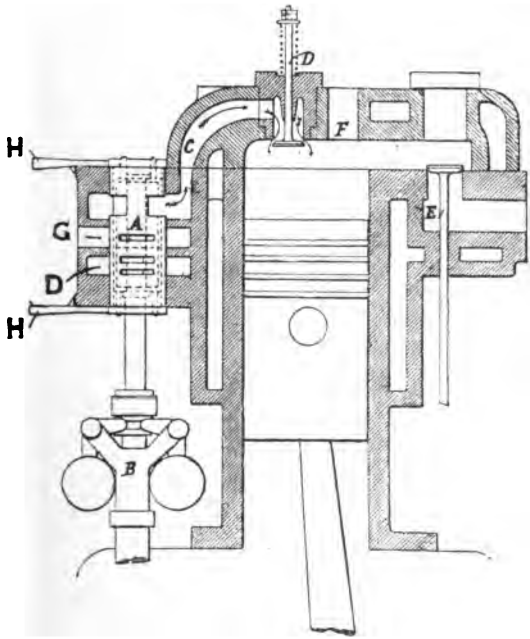


FIG 53.

by a cam on the shaft B. The current is supplied from an induction coil, and its direction of flow is easily reversible. In fact at each stop and start of the engine, the direction of the flow is automatically reversed by the action of switching off and on the coils. The object of this arrangement is to obviate the waste which necessarily takes place at the posi-

tive pole of the sparker by the abrasion of small particles of metal from the positive pole and by their deposit on the negative pole. This frequent reversal is said to preserve the sparker. Other points in design are worthy of notice—thus, for instance, the adjustment for the crank shaft bearings is effected by means of wedges placed beneath the brasses, thus taking up the downward wear and keeping the shaft alignment and cylinder clearance constant. Again, the closed oil chamber at the base of the engine serves to lubricate the piston and other working parts by the splash of oil from the connecting rod.

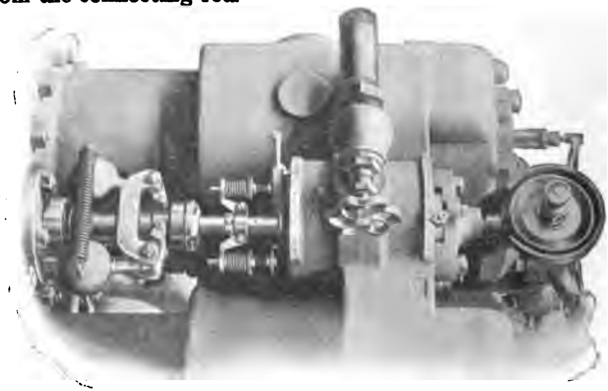


FIG. 54.

Fig. 53 shows the construction of the governor gear. Gas enters the annular space G and is drawn through the slot A on its way to the mixing chamber C. The annular space D is in communication with the outer air, and the two slots afford an inlet to the air on its way to the mixing chamber C. The slots for the admission of the air and gas are simultaneously opened or closed by the governor B—thus for any given volume of charge, the relative proportions of air to gas are always the same. It might, however, be desired to

change the proportion of air to gas. It should be remembered that a mixture of about ten volumes of air to one of gas is, when coal gas is being used, the most economical charge. If the engine were unavoidably to become overloaded for a short period of time it would be necessary, in order that the speed might be maintained, to increase the proportion of gas. This can be done in the "Westinghouse"

FIG. 55.

engine by moving the handles H, H so as to increase or diminish the length of the slots A. It will be seen that the movement of the governor affects the width of the slots through which the air and gas pass, while the length is independently moved by means of the handles. An outside view of this governor arranged for a two-cylinder engine is shown in fig. 54.

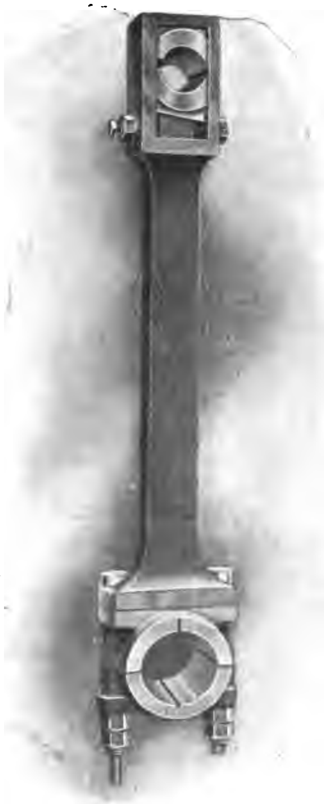


FIG. 56.

For the large sizes of engines a self-starter is provided. This consists of a compressed air chamber filled to a pressure of 160 lbs. per square inch, by means of an air pump driven by hand or belt as and when desired. In order to start the engine, the ordinary exhaust cam is put out of action and a supplementary cam brought into gear with the exhaust stem, which thereby lifts the exhaust valve at each upstroke of the engine piston. The inlet valve J, fig. 52, is closed and put out of action and a valve opened between the compressed

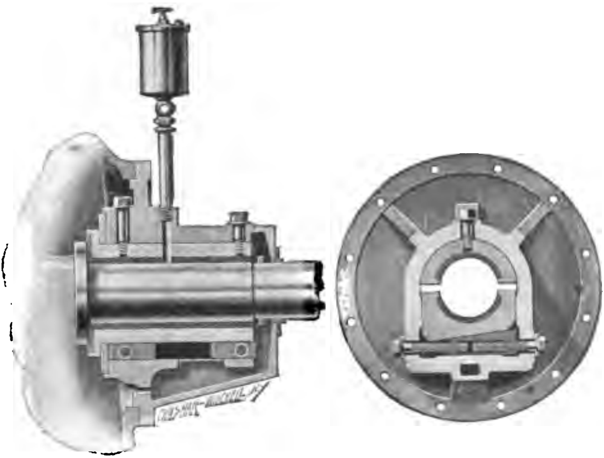


FIG. 57.

air cylinder and the engine. Compressed air is admitted to the engine cylinder through the controlling valve shown in fig. 55. The large engines manufactured by the Westinghouse Company are built with two or three cylinders and in such cases one of the cylinders is used as an air motor for starting, while the other cylinders are drawing in their charges of gas and air. When a few ignitions have taken place, the compressed air is turned off and the last cylinder put into action for receiving its charge of gas. These

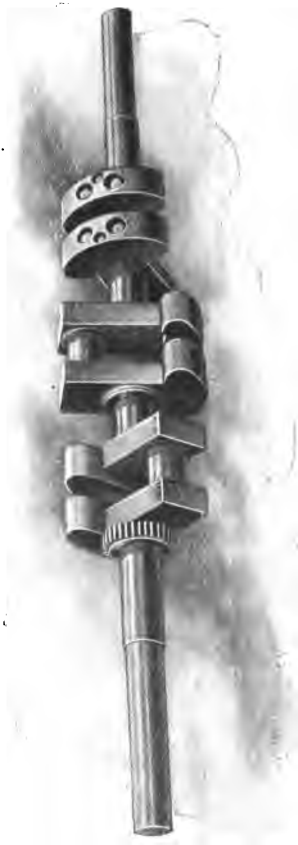


FIG. 58.

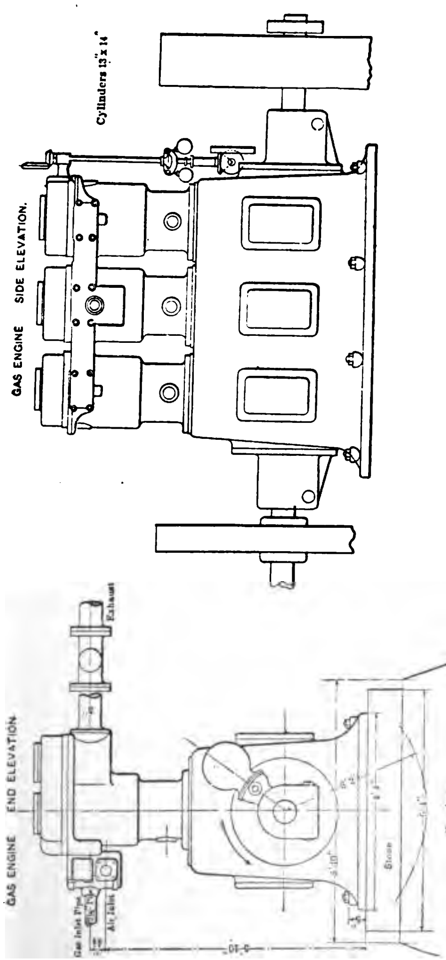

GAS ENGINE SIDE ELEVATION.
Cylinders 13" x 14"
GAS ENGINE END ELEVATION.
Exhaust
Air Inlet
Stone
Foundation

FIG 59.

engines are now being built in sizes ranging from 10 to 650 H.P. The company are now constructing the largest gas engines on record, namely, two three-cylinder engines of 1,000 brake horse power each.

Its external appearance is somewhat similar to the standard types of high speed steam engines now on the market, whilst internally there are points of resemblance. The general design of the connecting (fig. 56) follows along approved lines, means of adjustment for wear of the little end brasses is provided, whilst the big end brasses are of the type usually found in marine engine practice. The end bearings of the crank-shaft are shown in fig. 57, and they form part of the end covers of the crank chamber, similar to those found in the earlier patterns of the Willan's steam engine. The means of adjusting the brasses by wedges will be easily understood after examining the figure, whilst the true alignment of the bearing is assured by the machined flange. The crank shaft (fig. 58) is fitted with balance weights; the method of attachment by bolts is clearly shown.

CHAPTER VI.

SELF-STARTERS.

THE Forward engine is fitted with Lanchester's patent self-starting gear, the application of which is shown in the following drawing. Fig. 60 shows a general view of the arrangement fitted to the engine. Gas enters the combustion chamber from a ½ in. branch through the valve A. The stream of gas and air passes out of the cylinder through the cock C, and ignites, when rich enough in gas, at the naked flame shown at F. The mixture burns at the orifice, at first with a pale blue flame, which becomes deeper in colour as it becomes richer. When the flame burns with a roaring noise the valve A should be closed. The flame at the orifice will now strike back into the cylinder and ignite the entire volume of the mixture in the combustion chamber. The action of this starter depends entirely upon the size of

the outlet through the cock, for if this outlet were large in diameter the velocity of the gas would be less than the velocity of flame propagation ; consequently, immediately the gas ignited at the orifice, the flame would strike back and ignite the mixture in the cylinder before it was rich enough to give a starting impulse to the piston. This

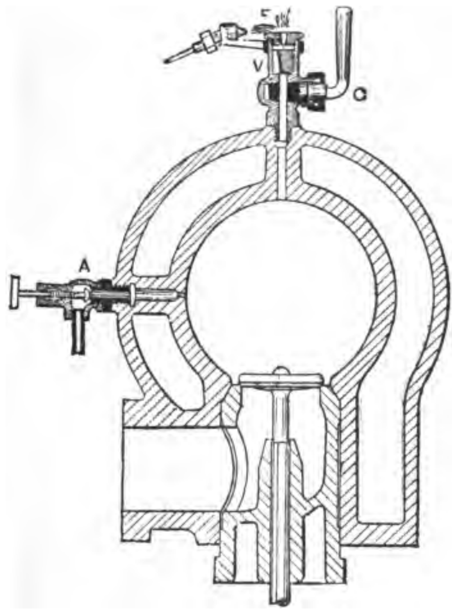


FIG. 60.—Lanchester's Self-starter for Gas Engines.

is prevented by the contracted area of the outlet nozzle, causing the gaseous mixture to flow rapidly from the orifice. This is, of course, the principle upon which all atmospheric burners depend, and the slight report which is always heard when turning out an atmospheric gas stove, is due to the return of the flame along the pipes towards the stop cock at the moment the latter is turned off.

To prevent the escape of the pressure through the cock from the engine cylinder, when the first ignition takes place, an automatic valve is placed inside the chamber in the cock C. This valve is shown at V, and rests upon its lower face, excepting when its weight is overcome by the

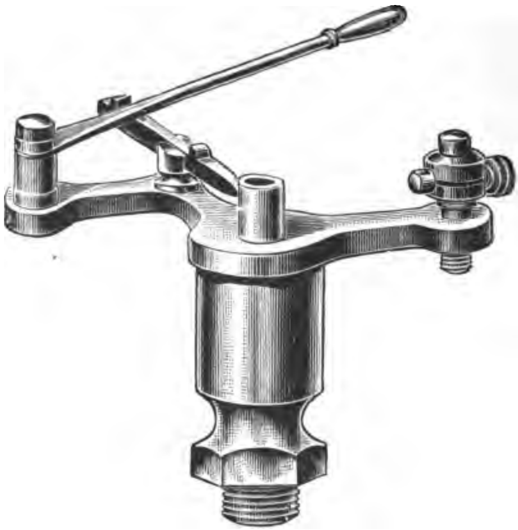


FIG. 61.—Green's Self-starter for Gas Engines.

rush of an explosion. Grooves cut in this valve allow the passage of gas through the open cock so long as the valve remains in its lowest position. Immediately the starting impulse is given this valve is projected upwards, and as the grooves are not deep, the centre of the valve entirely closes the orifice, thus preventing the escape of the pressure. The cock C may now be turned off. In using this starter the engine must always be barred round to the commencement of its *working* stroke, otherwise serious damage may be done to the gas bags and other fittings.

A self-starter similar in principle to that already described is shown in figs. 61 and 62. This is known as Green's self-starter. It differs from that already described in the construction of the automatic valve. Referring to fig. 62, the passage marked 17 communicates directly with the combustion chamber. The valve B is cylindrical, and hollowed out to the point C, but solid at the lower end. The holes 4_1, 4, when the valve is in position for starting, form a passage for the gas and air, which rise to the orifice 6, and ignite by the naked flame issuing from the gas burner E. When the mixture issuing from the orifice 6

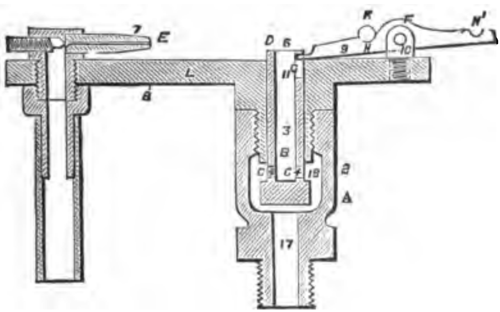


FIG. 62.—Green's Self-starter for Gas Engines.

burns with a dark blue flame, the gas supply to the combustion chamber is shut off, the flame strikes back through the passages 4, 18, 17, and ignites the charge in the combustion chamber. The pressure thus generated forces up the valve B, thereby closing the passages 4_1, 4. The hand lever shown in fig. 61 is made of spring steel, and when in the position shown presses downwards upon the lever engaging with the automatic valve, thus effectually keeping it in its highest position, and preventing the escape of pressure from the combustion chamber when the engine is working. When the starter is in use, the hand lever is

drawn forward, and holds the automatic valve in the position shown in fig. 62. The advantages of this form of construction are that the automatic valve serves to throw the starter out of use, and being at the same time accessible from the outside, there is no inconvenience occasioned by the valve sticking.

The self-starters we have described ignite the mixture at atmospheric pressure; hence the impulse of the starting stroke is limited, and the maximum pressure in the cylinder will not usually exceed 80 lb. per square inch. In order to increase this pressure, Mr. Dugald Clerk was the first to apply the apparatus shown diagrammatically in fig. 63, which is known as the Clerk pressure starter. A chamber C is connected to the engine cylinder by the pipe P and a check valve V. The passage H leads into C from a hand

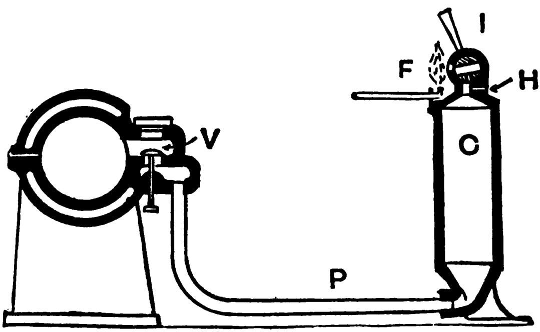


Fig. 63.—The Clerk Pressure Starter.

pump, which is used to charge C and the engine cylinder with an explosive mixture *slightly* above atmospheric pressure. I is an igniting valve, through which the mixture passes when open, and comes in contact with the flame F, which causes the ignition. The action of the starter is as follows: The cylinder and chamber C being filled at atmospheric pressure with an explosive mixture, the igniting

valve is opened, the mixture fires at the orifice, and the flame strikes back into the chamber C. As the gas burns downwards it expands, driving a large portion of the mixture yet unburnt through the pipe P into the cylinder. In this way the pressure of the mixture in the cylinder is considerably raised before the flame reaches it. When, therefore, the mixture in the cylinder ignites, it explodes

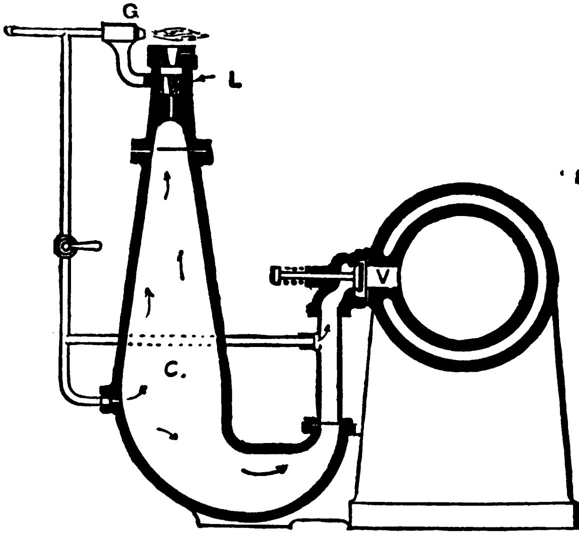


FIG. 64.—Clerk-Lanchester High-pressure Self-starter.

violently, developing a maximum pressure of 190 lb. per square inch, instead of 80 lb., as with the low-pressure starters of Lanchester and Green.

The self-starter shown in fig. 64 is known as the Clerk-Lanchester. It is, as will be seen, a combination of the Clerk pressure starter and the Lanchester automatic valve. It is otherwise more convenient than the Clerk starter, as no pump is required to fill the cylinder and chamber C with

an explosive mixture. The action of this starter is as follows: The engine cylinder and chamber C are filled with air at atmospheric pressure. To ensure pure air occupying these spaces, it is usual to open the valve V before the engine is stopped, thus drawing pure air through C into the cylinder. When ready for starting, the engine is barred round to the commencement of its working stroke, the valve V opened, and gas turned on; the gas jet G is lighted, and serves to ignite the mixture as it issues from the orifice. Immediately the flame strikes back into the chamber C the valve L lifts and closes the orifice. The action is now precisely the same as already described in the Clerk pressure starter.

All makers supply with their larger engines some self-starter, which, though differing in detail from those already described, may nevertheless be recognised as dependent upon one or other of the principles explained.*

CHAPTER VII.

Two-cycle and other Engines.

The large number of gas-engine makers who are now competing has given rise to a variety of distinguishing names, which the author ventures to think are far in excess of the corresponding differences which those names are supposed to characterise. Having now fully described a variety of engines which may be considered typical, but little will be gained by dwelling at length upon minute details. It will be advisable, before leaving the descriptive side of the subject, to touch briefly upon points of interest in connection with engines not already described.

The Day gas engine, made by Messrs. Day and Co., Bath, is illustrated in fig. 65. This engine is noticeable for its simplicity and fewness of parts. Though not suitable for

* Other methods of starting gas engines are described on pages 41, 52, 54, 56, 61, and 67

large powers, it is a moderately efficient engine where only light work is done. It will be noticed that the crank and connecting rod run in a closed space, into which is fitted one mushroom valve. Through this valve both gas and air are

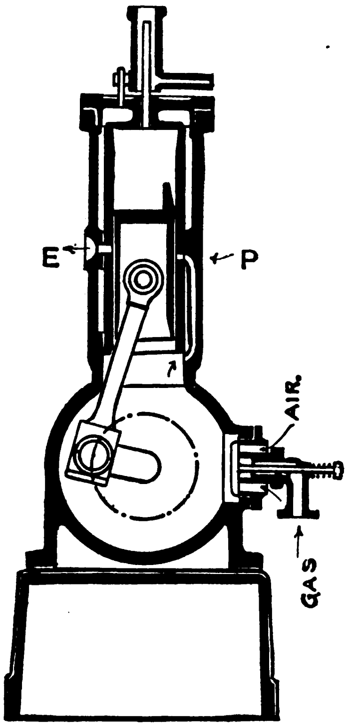


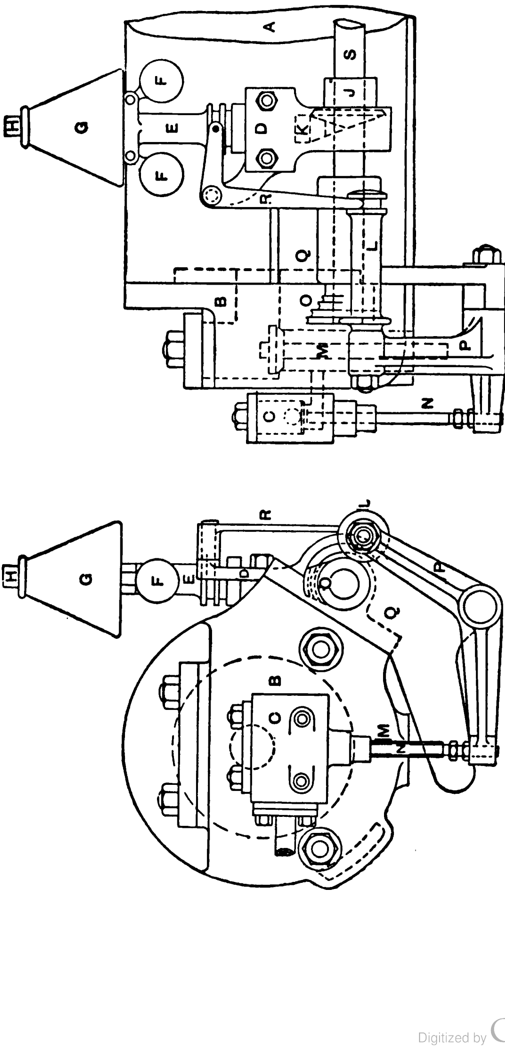
FIG. 65.—The Day Gas Engine.

sucked during the up stroke of the piston. On the down stroke this mixture is slightly compressed. When the piston descends, the port P is uncovered, and the mixture flows from the lower chamber into the working cylinder.

On the return stroke the port entrance to the cylinder is covered by the piston, and the charge being compressed into the ignition tube I, is fired without the use of a timing valve. The piston is moved downwards by the expanding gases, which escape through the port E when uncovered by the piston. It will be observed that the exhaust port is uncovered before the inlet, and therefore the pressure of the expanding gases is released before the end of the stroke. The inlet and exhaust are open together for a short period of time, and the first rush of fresh mixture is deflected upwards into the cylinder; in this way the greater portion of the burnt products are displaced. As the working of the valve is entirely automatic, the engine may be run in either direction without any alteration whatever. This engine gives an impulse every revolution.

The Palatine gas engine is constructed somewhat upon this principle, the crank shaft running in a closed chamber. The object of this design is to supply a scavenger stroke, and the impulse only occurs every two revolutions. The objection to these designs is the inaccessibility of the connecting rod.

Messrs. Alfred Dougill and Co., Leeds, make an Otto cycle engine, which is noticeable for its patent governor gear. This gear is shown in figs. 66 and 67. The object of the arrangement is to keep the ratio of air to gas uniform throughout all variations of load. Instead of merely intercepting the gas supply, the governor acts on both the air and gas valves, and when the speed rises the total volume of the charge admitted to the cylinder is decreased, but the ratio of air to gas remains unchanged. The essential parts of the gear are the stepped cam O and the sleeve L. The lever P lifts both the gas and air valve spindles N and M, and is driven by the stepped cam O operating upon the movable sleeve L. In fig. 67, L is shown in line with the highest step; consequently the admission of the charge is a maximum. When the governor balls F rise the sleeve is drawn towards the lower steps, and



FIGS. 66 AND 67.—DOUGILL'S GOVERNOR.

the lift of the valves is diminished. It will be observed that since an increase in speed must take place on the explosion stroke, the roller of the sleeve L is at the time of the increase running upon the cylindrical portion of the cam O. For this reason it is perfectly free to move sideways when the speed varies. Diagrams obtained from this engine are shown in fig. 68. From these it will be seen that the compression is reduced when running light, and the explosion is nearly as rapid as at full load. This governor is said to prevent a variation in speed of more than 2 per cent on account of the continuity of the impulses even at

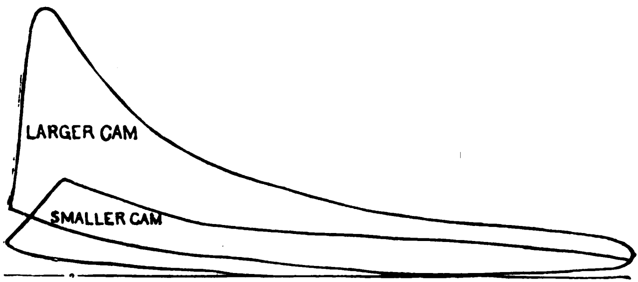


FIG. 68.—Diagram taken from Dougill's Otto Gas Engine.

light loads, and it is therefore highly suitable for electric lighting purposes.

The Acme engine, made by Messrs. Alexander Burt, Glasgow, was designed with the object of obtaining maximum expansion. This engine comprised two cylinders, working with two crank shafts. The shafts were geared together with two to one gearing, so that one piston made twice the number of strokes of the other. The gases expanded from the higher speed cylinder to the lower one, and the exhaust was said to be much cooler than usual. The results obtained showed the engine to be economical; but the noise of the gearing, and the extra wear of these parts, probably operate against the more general adoption of the engine.

During the last few years there has been a growing tendency for all makers to adopt the Otto cycle in preference to their own particular makes. We have already mentioned that this cycle leaves much to be desired in regard to steadiness of running. It is not surprising, therefore, that a reaction is taking place, and not a few are trying to solve the problem of constructing a satisfactory engine having an impulse every revolution. The Griffin engine, as already described, gives an average of one impulse every revolution, but utilises both ends of the cylinder for the combustion of the charges. For more reasons than one it is desirable to utilise only the space at the back of the piston as an explosion chamber, the forward end merely acting as a pump.

The first engine built, in 1878, to give an impulse every revolution was designed by Mr. Dugald Clerk, Assoc.M.Inst.C.E. It will be well to describe briefly the working of this and other two-cycle engines, not because they are regarded as modern gas engines, but because the author believes that the gas engine of the future will probably be a combination of past and present attempts to construct an engine having an impulse every revolution.

Mr. Dugald Clerk's first engine consisted of two cylinders, one of which was used as a motor cylinder, the other as a pump to deliver the mixture into a reservoir at a pressure of about 70 lb. per square inch. The motor piston travelled nearly to the end of the cylinder. The usual combustion chamber was dispensed with. When the motor piston had moved forward about 2 in. on its working stroke, a slide valve admitted the mixture from the reservoir. The supply being cut off, ignition took place. In this engine the inventor sought to combine the advantages of compression before ignition with the complete expulsion of the exhaust gases by minimising the clearance at the back of the cylinder. The difficulties met with in this engine were chiefly due to back ignition in the compression reservoir, and to excessive shock in the motor cylinder. The former of these diffi-

culties was never overcome, though the latter was minimised by modifying the shape of the combustion chamber.

In 1880 Mr. Clerk designed another engine, in which the pump delivered its charge directly into the motor cylinder slightly above atmospheric pressure. The exhaust ports were uncovered by the working piston, and just before being closed by the return of the piston the displacer pump delivered its charge into the working cylinder. This was compressed on the return stroke and ignited at the commencement of the forward stroke. Many engines of this pattern were made, and in the larger sizes worked quite as economically as the best engines of their time, though there were difficulties in the governing. It is also obvious that there was great risk of delivering the unburnt mixture into the

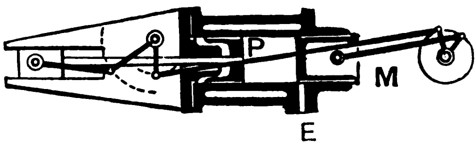


FIG. 69.—The Clerk Two-cycle Engine.

exhaust ports, and Mr. Clerk states that in the smaller engines this could hardly be avoided.

In 1890 Mr. Clerk built another engine, which is shown diagrammatically in fig. 69. Here M is the motor piston, P the charging and expelling piston. The link work is so arranged as to give a maximum speed of motion to one piston whilst the other remains almost stationary. Thus, when M uncovers the exhaust E, P travels rapidly towards it and expels the burnt gases; at the same time P is drawing a fresh charge into the back of the cylinder. The two pistons then travel back together, and the charge passes from the back of piston P to the space between the pistons. Upon ignition, M moves forward, whilst P remains practically stationary. The inventor believes that this engine is well adapted for both moderate and large

powers, and that it will give exceptionally economical results when given a thorough trial. It seems hardly probable that its mechanical efficiency will be high after a short period of use, though the engine is no doubt very effective in design from a thermal, if not from a mechanical point of view.

Messrs. Robey and Co., of Lincoln, have for some years manufactured an engine upon the Otto cycle, with improvements introduced by Messrs. Richardson and Norris. An important feature of the engine is the ease with which it can be reversed. The cams are made to run in either direction, so that the alteration can be effected in a few moments. The speed is regulated by Richardson's high-speed governor, similar to that used on other Robey engines, fully described and illustrated in the *Engineer* of October 14th, 1892. The governor is loaded partly by dead weight and partly by springs, which latter can be regulated whilst running to give the required speed. The governor acts upon the gas valve, cutting off the supply of gas when the speed increases. Many of the Robey engines are specially designed for electric lighting, and for this purpose Mr. Richardson has introduced an electrical governor. This, in principle, consists of a solenoid wound as a shunt from the current generated by the engine and dynamo. A soft iron core is sucked into the solenoid as the voltage rises. Its movement is connected by a bell-crank lever to the pecker operating the gas valve, causing it to hit or miss the knife edge on the gas-valve spindle. By this means the gas supply may be entirely cut out.

Mr. Norris is now perfecting a two-cycle engine, with the object of increasing the steadiness of running and increasing the power derived from a given weight of engine. The arrangement is briefly as follows: The front end of the cylinder is closed, and acts as a pump for drawing in the charge and expelling it to the combustion chamber at the back of the cylinder. The piston rod carries a slight enlargement upon it, which traverses a cylinder attached

to the front cover. This small cylinder is used to pump in the gas.

The cycle of this engine is carried out in the following order: The return stroke of the piston draws air into the front end of the cylinder through an automatic valve, whilst gas is drawn into the small cylinder attached to the inside of the front cover. The next outstroke compresses the air into an intermediate chamber at about 5 lb. per square inch above atmospheric pressure, until the piston has completed nine-tenths of its outward stroke. At this point a port in the engine piston connects the combustion chamber with the receiver. The exhaust valve is opened a little before nine-tenths of the outward stroke, and a column of exhaust gases is therefore already in motion by the time air enters the combustion chamber from the receiver. Air continues to flow from the receiver through the combustion chamber until one-tenth of the inward stroke is complete, when the exhaust valve closes. At this time pure air only occupies nine-tenths of the cylinder volume. After the air and exhaust valves are closed, gas from the small pump is admitted to form an explosive mixture, which is compressed during the remainder of the instroke. The compression, in the opinion of Mr. Norris, should not exceed 95 lb. per square inch.

In larger sizes of this engine a differential piston is used, in order to increase the volume of air pumped into the receiver.

CHAPTER VIII.

French Engines.

SIMPLEX GAS ENGINE.

The Simplex gas engine was patented by Messrs. Edward Delamare and Malandin, in 1884. Whilst resembling in many respects the Otto engine, this motor has several distinctive features. It works upon the usual Otto cycle, but the charge is fired just after the commencement of the outstroke of the piston, instead of, as usual, on the dead centre. It is stated that this modification greatly reduces shock upon the working parts. The ignition of the charge is effected by means of an electric spark produced by a battery and induction coil. The difficulty usually met with in electric igniters is found to be due to the inefficiency of the *make and brake* mechanism. This is obviated in the Simplex engine by the generation of a continuous spark within a specially-formed chamber. The entrance to this chamber is controlled by a slide valve, and the charge in the cylinder can only come in contact with the continuous spark when the port through the slide opens communication between the cylinder and sparking chamber. Many of the continental engines fire by electricity, but this method has been little used in this country. There are points in favour of electric ignition; but the maintenance of the battery, the possibility of the sparking points becoming coated with deposit, and the difficulty of maintaining the insulation are serious objections, and it is probable that as the hot tube is made more durable it will entirely supersede electric ignition.

The governing of this engine is effected by the use of an air cylinder and piston. The piston is stationary, and the air cylinder moves backwards and forwards with the igniting slide spindle. Air enters the cylinder through an orifice regulated by a micrometer screw. If the speed of

the engine increases, the air in the cylinder is unduly compressed, and so forces outwards a second piston, which is in communication with the air cylinder. This latter piston carries the knife edge for actuating the gas valve. The micrometer screw can be so regulated as to cause the outward motion of the second piston at any desired maximum speed. In this way the knife edge is moved out of line with that on the gas-valve spindle, and the entire supply of gas is cut off. On larger engines of the Simplex pattern a form of pendulum governor has been adopted.

The self-starter fitted to the Simplex engine consists of a three-way cock, suitably proportioned to admit a charge of gas and air to the cylinder. The cock is opened when the piston is at the commencement of its forward expansion stroke, and when charged the current is turned on. The high temperature of the electric spark renders the ignition of a weak mixture certain.

LENOIR ENGINE.

This is another engine in which electric ignition is used to fire the charge. The chief object of the designer was to obtain a very high compression and temperature of the charge before firing. This was effected by jacketing the working part of the cylinder only. The compression chamber was bolted to the cylinder casting, with an asbestos joint to prevent the transmission of heat from the compression chamber to the jacketed cylinder. The compression chamber was cast with a series of deep ribs outside, to distribute the heat by convection of air instead of by the use of a water jacket. By this means the combustion chamber was kept at a much higher temperature than the working cylinder, and a weak mixture was easily ignited by the electric spark.

CHARON ENGINE.

An attempt is made in this design of engine to expand the charge beyond its volume at atmospheric pressure when drawn into the cylinder. To effect this, the air admission

valve remains open during part of the compression stroke, and through it a portion of the charge is driven into a long spiral tube open to the atmosphere, and of sufficient length to retain the mixture forced into it. The valve then closes, and the charge is compressed. After expansion and exhaust the gases in the spiral pipe are drawn into the cylinder with the new charge, and the cycle is repeated. The engine is governed by reducing the gas supply and by allowing more of the charge to enter the spiral pipe, thus reducing the quality and volume of the gas ignited.

NIEL ENGINE.

In this engine the charge is admitted through a conical revolving plug, which is kept tight during the compression and explosion strokes by the pressure generated in the cylinder, acting upon the back of the plug so as to force it into its seating.

LALBIN ENGINE.

This engine is designed with the object of equalising the turning effort upon the crank shaft. It consists of three cylinders, one placed vertically with the crank shaft beneath it, and one at each side inclined at an angle of 60 deg. with the horizontal. Each cylinder works upon the Otto cycle; hence in two revolutions the crank shaft receives three impulses, instead of only one. The mechanism is so arranged as to permit of the engine being reversed.

German Engines.

BENZ ENGINE.

Of the German engines perhaps the Benz is one of the most interesting from a constructional point of view, inasmuch as the cylinder is arranged to give an impulse every revolution, without the aid of a separate large pump, such as used upon the early gas engines of the Clerk type. The engine is horizontal, and has electric ignition. The cylinder is closed at both ends; the back of the piston receives the

impulse from the ignition of the mixture, and the front of the piston drives, at each forward stroke, a volume of fresh air through a suitable valve into an air reservoir formed in the bed plate of the engine. A small plunger, worked from the crosshead, acts as a gas pump, and the plunger cylinder becomes filled with gas during every forward stroke of the engine. It is possible with this arrangement to produce an explosion at the commencement of each outstroke of the piston. Imagine the gases to have ignited and driven the piston forward. By this stroke the air reservoir is replenished and the gas pump charged. On the return stroke the exhaust valve opens, and just before the end of the stroke the air from the reservoir is admitted to the combustion chamber. This effectually clears out the burnt gases and replaces them with fresh air, which at the same time tends to cool the cylinder. The air and exhaust valves then close, and the gas pump forces its charge into the combustion chamber. The air and gas mingle sufficiently to permit of ignition by means of the electric spark. The consumption of gas in these engines is said to be about 23 cubic feet per indicated horse power.

KOERTING-LIECKFELDT ENGINE.

The first of these engines was designed with vertical cylinders, and is specially interesting for the method of ignition of the charge. By a special apparatus, shown in fig. 70, the external flame F ignites the charge. The chamber C is in direct communication with the motor cylinder during the compression stroke and at the time of ignition. The cylindrical piece B is free to move upwards under the pressure of the mixture. The rod R is worked up and down by a lever, thus opening and closing the port P opposite the naked flame. When compression commences the rod R is raised, so that the port P is open to the flame. The mixture enters the chamber C, and raises the plug B. It is then forced through the small entrance of the cone, and, expanding, arrives at P at nearly atmospheric pressure, where it is

ignited by the flame F. The flame then travels a little way down towards the small end of the cone, but cannot get very far, because the velocity of the outcoming gas is greater at the lower end of the cone than the rate of flame propagation. The rod R descends at the end of the instroke of the piston, and, covering the port P, effectually stops the flow of gas; the pressure within the cone becomes equalised, the

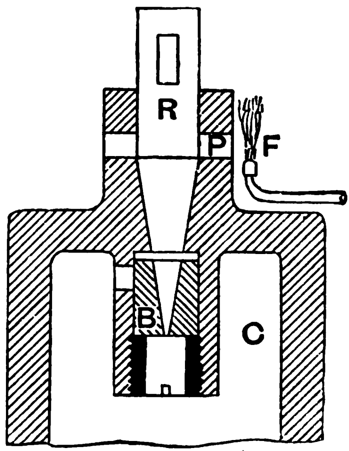


FIG. 70.—Koerting-Lieckfeldt Igniter.

piece B drops, and the flame travels downwards towards the chamber C, and so ignites the entire charge.

The governing is effected by a somewhat novel arrangement of the centrifugal type, which cuts out the gas and air supply, and at the same time keeps the exhaust open, so that the products of combustion are drawn back into the cylinder. This method is advantageous in oil motors, inasmuch as it helps to maintain the temperature of the combustion chamber; but it is a doubtful practice in connection with gas engines.

THE DAIMLER GAS ENGINE.

This engine was first exhibited in 1889, and is specially noticeable for its high speed and simplicity of parts. The later designs have two cylinders, placed nearly vertically over one crank shaft. The connecting rods work upon one crank pin, set between heavy balanced crank webs. The connecting rods and crank are entirely encased in an air-tight covering, in a similar way to the engine shown in fig. 65. An automatic air valve admits air only to this casing as the pistons rise. The gas and exhaust valves are placed at the upper end of each cylinder, the gas valve being automatic, whilst the exhaust is lifted by the equivalent of a cam on the disc crank web. Instead of the air port P, shown in fig. 65, a valve is placed in the centre of each piston, for passing the air from the crank chamber to the combustion chamber. The action of the engine is as follows: Upon the down stroke one cylinder expands its burning charge, whilst the other draws in gas, and at the same time passes air from the crank casing into the working cylinder through the valve in the piston. On the return stroke the one cylinder exhausts, whilst the other compresses. In this way an impulse is received every revolution, although each individual cylinder works upon the Otto cycle. The governing is effected by holding open the exhaust valve every revolution when the speed rises, thus preventing the automatic gas valve from being opened by the formation of a slight vacuum in the cylinder. This engine is worked entirely without jacket water. This is no doubt possible on account of the high speed at which it is run (from 500 to 700 revolutions per minute), coupled with the fact that only very small powers are attempted. It is claimed that the engine will consume about 33 cubic feet of gas per hour per I.H.P., and when it is remembered that the engines only range from 1 to 2 horse power, this figure is not very excessive.

THE CAPITAINE GAS ENGINE.

The Capitaine gas engine is designed to give a high speed of revolution with a nominal piston speed. This is done by increasing the diameter of the cylinder, whilst decreasing the length of stroke. The combustible mixture is drawn into the cylinder through an enlarging entrance, so that its velocity is decreased, the object being to prevent the mingling of the new charge with the exhaust gases. How far this arrangement is effectual it is difficult to estimate, though it is certain that the time of combustion is lengthened by the presence of burnt products in a rich mixture of, say, eight volumes of air to one of gas. The ignition tube in the Capitaine engine is of porcelain, and is claimed to be very durable and efficient.

THE OECHELHAUSER TWO-CYCLE ENGINE.

The Oechelhauser gas engine is a simple form of two-cycle motor. The engine was designed with a view to securing very high compression, so that ignition might be made certain when using gases derived from iron furnaces. Referring to fig. 70A, it will be seen that two pistons work in one long cylinder. The left-hand piston is connected direct to the crank shaft, whilst the right-hand piston is connected to a crosshead, attached to the crank shaft by means of side return rods terminating in the usual form of connecting rod. The outside crank pins are placed diametrically opposite the central one; consequently the two pistons retire simultaneously towards the centre line of the cylinder until they nearly touch, thereby compressing the mixture within the cylinder. When ignition takes place the two pistons simultaneously move apart. One piston uncovers the exhaust ports as it completes its outward stroke, and at the same time the other piston uncovers first an air port through which a draught of air enters the cylinder and effectually clears it of the products of combustion. After admitting the air the same piston completes its outward

stroke, and in so doing uncovers a second set of ports which admit a mixture of air and gas. The mixture of air and gas and the scavenging charge of air are forced into the cylinder by means of a pump. The right-hand side of the pump plunger is in contact with air and gas, while the left-hand side discharges air only for the purpose of scavenging. It will be observed that the engine has no valves to the cylinder, and in this respect is similar to the Trent gas engine now out of date. The difficulty hitherto experienced with this arrangement of gas admission has been the escape of the fresh charge thrcugh the exhaust port. In the engine

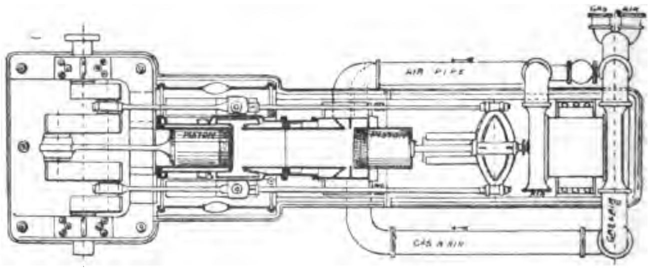


FIG. 70A.

under discussion it is stated that the volume of explosive mixture admitted to the cylinder is only 70 per cent. of the volume of the cylinder, but this is the only safeguard against loss of gas through the exhaust. It is probable that with a long exhaust pipe some of the fresh charge will find its way therein. As, however, the engine was designed with a view to utilising the waste gases from iron furnaces this loss is not serious, and the engine may in spite of this defect have a wide field of usefulness.

An impulse is, of course, received every revolution, and the dimensions of the engine for a given power are therefore

considerably less than usual. Thus an engine developing 500 B.H.P. has a cylinder diameter of 26·6 in., and an engine of 1,000 B.H.P. has a diameter of only 36·8 in. The engine is governed by reducing the quality of the mixture drawn into the pump by throttling the gas admission valve G. At the same time the valve V is opened by the governor, and allows some of the compressed mixture of gas and air to return to the suction pipe of the pump.

Some of these engines are now at work at the Horde Ironworks, Westphalia.

CHAPTER IX.

Testing Gas Engines.

The testing of gas engines specially fitted up for experimental work is a comparatively easy task, and involves but little forethought from the experimenter; when, however, complete tests are to be made by the use of improptu apparatus, there is ample scope for the ready exercise of one's wits in arranging simple and reliable means of measuring the quantities to be dealt with. In what follows we shall confine our attention to testing a gas engine alone—that is, not in conjunction with the machinery it may be driving. There may be cases in which the work done by the driven machine is easily ascertained. The horse power obtained from the terminals of a dynamo driven by a gas engine affords a measure of the useful work done, and, making allowance for the efficiency of the dynamo, gives the work transmitted from the engine flywheel to the dynamo pulley. The work done in pumping a certain weight of

water against a known head, in a given time, is easily converted into useful horse power. But in all such cases the efficiency of the machines has to be allowed for, and as this may be a rather uncertain quantity, the only reliable method of testing is to run the engine separately, and to measure the power derived from the flywheels by suitable means.

The simplest and most reliable form of absorption dynamometer is illustrated in fig. 71. It consists of two or more

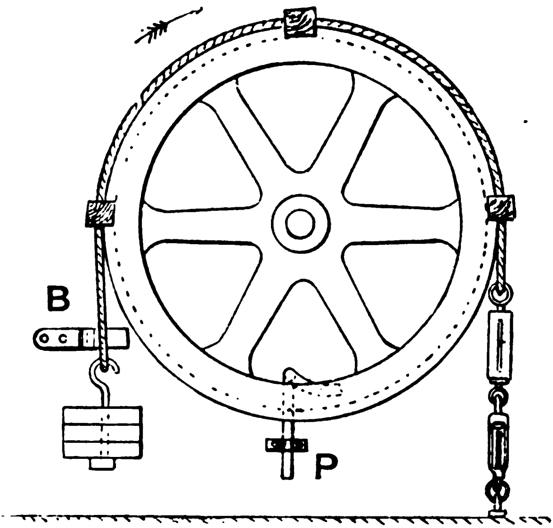


FIG. 71.—Friction Brake.

lengths of good hemp rope placed over the upper semi-circumference of the flywheel, and guided by wood blocks. The ends of the ropes are spliced together so that a spring balance may be attached at one side of the wheel, whilst a hook for carrying weights may be attached to the other side. The spring balance is anchored to the ground by

means of a hook and adjustable screw. The latter is of great convenience, for without means of adjustment the ropes often stretch and allow the weights to touch the ground. The weights for loading the brake should be cast with slots in, so that they may be placed upon the hook. When putting on the weights it is important that the slots should *not* coincide, for if they do the vibration will invariably cause the weights to fall over.

As a precaution against accident, the bar shown at B, fig 71, should be fixed to a suitable place and pass between the brake ropes. This prevents the undue lifting of the weights should the friction increase, and has often been the means of preventing a serious accident.

When absorbing large powers the rim of the wheel will become much heated, and its undue expansion may lead to fracture. It is therefore better to cast the rim in the form of a trough ($\sqcup$), and arrange for the delivery of cold water to the rim. When the wheel is revolving the water will follow it in the direction of rotation and completely line the trough; centrifugal force, acting upon the water, will prevent its displacement from the trough. In order to change the water in the rim the pipe shown at P, fig. 71, is fixed so as to skim the surface and drain away the water so caught. In this way a constant circulation may be maintained.

It is important that the brake weights be removed when the engine is being stopped, for if not the spring balance will be broken by their sudden fall. Another arrangement of the brake ropes is shown in fig. 72. This is more suitable for large powers, but whether one form or other is adopted is largely decided by convenience of arrangement, though for very small powers this latter form of brake is unsuitable.

When an economy trial is to be made the brake should be loaded, with the gas cock fully open, until the engine runs at its nominal speed, and fires every cycle with a full charge of gas. To adjust the brake and other measuring gear, a preliminary trial is invariably necessary.

In calculating the load upon the brake, the weight of the hook supporting the load should, of course, be included, and allowance should be made for any unbalanced part of the brake rope. The spring balance should always be tested before use, as the spring often gets a slight permanent set, and the index does not usually read *zero* when the balance is unloaded. It is of the highest importance that the weights hang freely, otherwise serious errors may arise. Although these remarks deal with details apparently trivial, we cannot too strongly urge the necessity of their strict

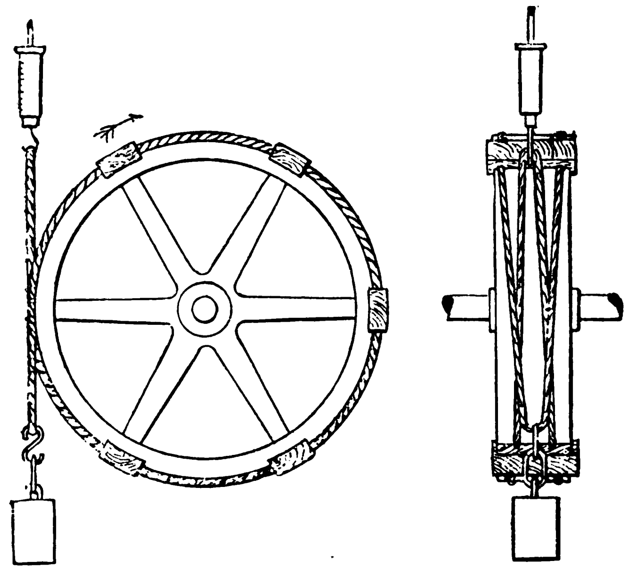


FIG. 72.—Friction Brake.

observance. A successful trial depends entirely upon the immediate observation of any disturbance of the measuring apparatus, and this can only be detected by constant watching.

During the trial the reading of the spring balance should be taken every five or ten minutes, according to the duration of the test, and from these an average may be calculated.

Let S = the mean spring balance reading in pounds;
W = the *total* weight hanging upon the brake in pounds;
R = effective radius of brake wheel in feet,
= radius of brake wheel + radius of brake rope;
N = revolutions of wheel per minute;

then the brake horse power

$$= (W - S)\frac{2\,\pi\,R\,N}{33000}.$$

For the same brake gear the factors

$$\frac{2\,\pi\,R}{33000}$$

remain constant; hence they may be worked out once for all, and the result be booked as the *brake constant* = C. The formula for the B.H.P. may then be written

$$\text{B.H.P.} = (W - S)\,N\,C.$$

Indicating.

Indicator diagrams obtained from gas engines cannot be entirely relied upon, though they may, with careful manipulation and a suitable instrument, be considered accurate enough for practical purposes. Before any important trial is to be made the indicator springs should themselves be tested under the conditions of temperature likely to occur. The average difference in the deflections of indicator springs when hot and cold amounts to as much as 5 per cent, and more in some cases; a cold test is therefore no guarantee. In the case of a gas engine, a simple method of testing the springs is as follows: Construct a stirrup of brass, as shown in fig. 73, having a hole in it sufficiently large to allow the indicator piston rod to pass

through it. This stirrup is placed in position by disconnecting the links attached to the pencil lever Next obtain a 100 lb. spring balance, and carefully test it, with the stirrup attached, against standard weights, noting the plus or minus errors, if any. Then pass the cord attached to the

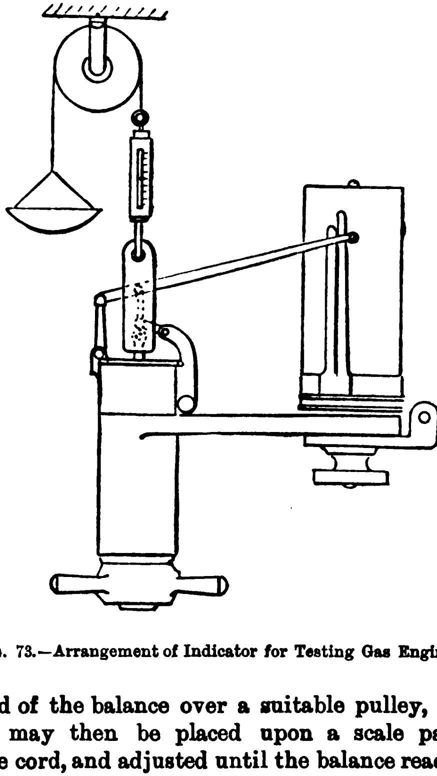


FIG. 73.—Arrangement of Indicator for Testing Gas Engines.

upper end of the balance over a suitable pulley, as shown. Weights may then be placed upon a scale pan at the end of the cord, and adjusted until the balance reads various definite loads. The use of an accurate balance can hardly be dispensed with, as otherwise the friction of the pulley will give an error of probably 5 per cent. By the arrangement

shown this is eliminated. Suppose a spring is being tested which is supposed to register 100 lb. per square inch for every one inch vertical movement of the pencil on the drum. Put the indicator on the engine, and arrange the tackle above it as shown in fig. 73. Start the engine, and whilst running *slightly* turn on the indicator cock so as to heat up the spring. When the indicator is at its working temperature, close the indicator cock and mark off the atmospheric line on the drum, revolving the latter by hand. Now load the balance until it reads 10 lb., and mark off the corresponding height of the indicator pencil. Proceed in this way by increments of 10 lb. at a time until the limit of the instrument is reached. Carefully measure the diameter of the indicator piston and obtain the area. In our example, suppose the area of the piston to be 0·5 square inch, and suppose when the balance reads 50 lb. the movement of the pencil above the atmospheric line measures 1·03 in. The pressure per square inch at 1·03 in. rise

$$= 100 \times 1{\cdot}03 = 103 \text{ lb. per square inch.}$$

But the actual pressure upon an area of 0·5 square inch is 50 lb., which is equivalent to

$$\frac{50}{0{\cdot}5} = 100 \text{ lb. per square inch.}$$

We see, therefore, that the indicator is recording a pressure 3 per cent in excess of the real pressure acting upon the piston.

When testing springs by this method, care must be taken not to overload the instrument, for it must be remembered that the whole stress is transmitted through the small piston rod of the indicator.

CHAPTER X.

INDICATORS FOR GAS ENGINES.

CROSBY INDICATOR.

THIS instrument, shown in figs. 74 and 75, has a well-earned reputation of being both durable and accurate at high speeds. The heavy stresses brought to bear upon the working parts of the instrument when used for gas-engine indicating has led to modifications in design, rendering it in every way more adaptable to gas-engine work. The most obvious alteration is the adoption of the straight link in the parallel motion, in place of a longer and curved link attached to the extreme end of the pencil bar. The indicator shown in fig. 75 is specially designed for the gas engine, but

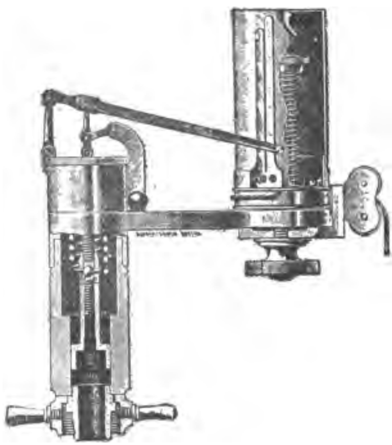


FIG. 74.—Crosby Indicator.

it can be used with equal efficiency upon a steam-engine cylinder. It will be observed that the barrel of the indicator is bored out to two diameters, the larger part

having an area equal to twice that of the smaller. Two pistons are supplied, to either of which the same spring may be fitted. In this way a spring marked 100 will register 100 lb. per square inch for each inch of vertical movement of the pencil when the spring is fitted on the *larger* piston. If, however, this spring is fitted to the *smaller* piston, as recommended for gas-engine work, then

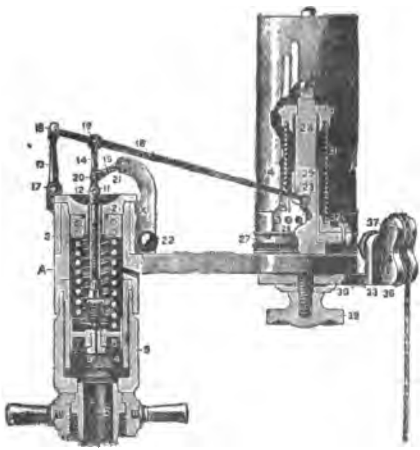


FIG. 75.—Crosby Indicator.

1 in. vertical movement of the pencil corresponds to 200 lb. pressure per square inch. With this instrument a variety of pressures may be dealt with by a small assortment of springs, a feature which readily recommends itself to the economist. In both designs the working cylinder is jacketed, to enable it to expand readily with the piston, and so obviate undue friction. All the piston springs are double spiral, to prevent the slight rotation and tilting of the piston-which is inevitable with a single spiral. A very light pencil bar is used upon these indicators for steam-engine work, but for gas-engine work a bar of H section is necessary to withstand the sudden jar due to the explosion.

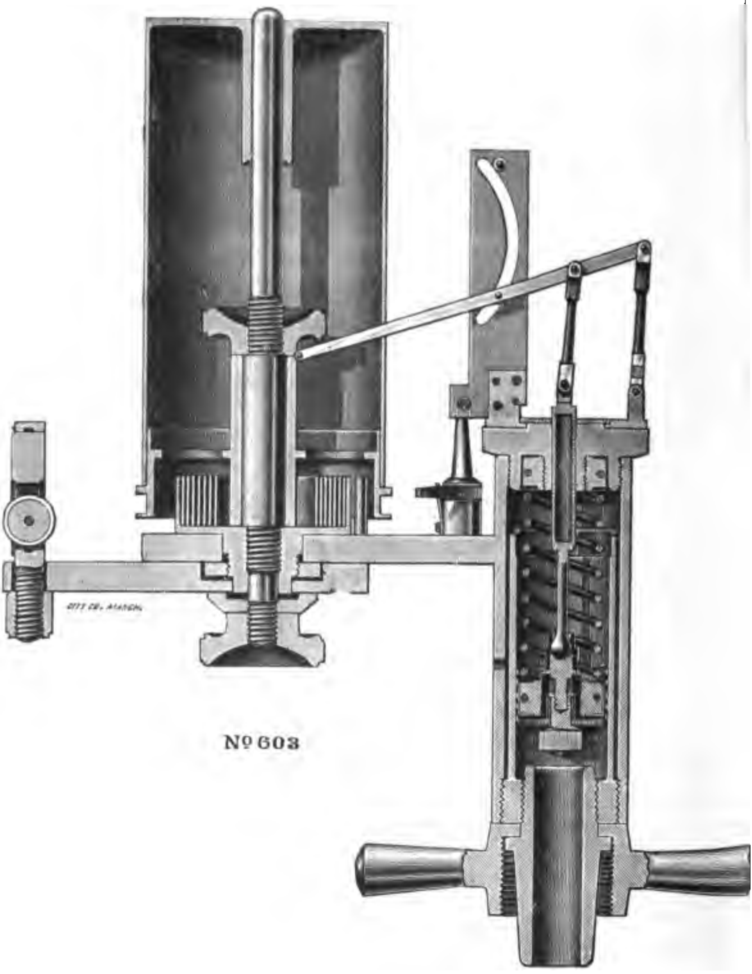


Fig. 76.—Section of Tabor Indicator.

The general construction of the indicators will be gathered from the illustrations.

THE TABOR INDICATOR.

The Tabor indicator is shown in fig. 76. Its three most characteristic features are the straight-line motion, the working cylinder, and the involute spring used to actuate the drum. The vertical movement of the marking point is secured by the curved slot, in which runs a small roller attached to the pencil bar. The cylinder in which the indicator piston works consists of a thin tube attached to the body of the indicator at its lower end only. The air jacket around the tube admits of free expansion of the tube when in use, and the thin walls of the latter assist in equalising the temperature. These indicators are somewhat larger than the Crosby, and will give diagrams 5 in. long at slow speeds, though when working at 200 revolutions per minute the makers recommend a 4 in. diagram, reducing it to 2 in. at 500 revolutions in order to minimise the effects of inertia of the drum.

The indicator illustrated in fig. 77 requires no reducing motion as usually fitted. It will be seen that the worm R engages with a worm wheel upon the drum B. The cord from the pulley O is attached to the crosshead of the engine, and travels through its entire stroke. Various lengths of diagrams are obtained by altering the size of the pulley O. The cage *d* contains the spring which gives the return motion to the pulley. The backlash of the worm gear is taken up by a spring within the drum B. By a simple movement of the milled head, shown at *u*, the pulley O disengages with the worm, and revolves freely upon its spindle. The drum B then remains stationary. This gear is very convenient for indicating at high speeds; but it is obvious that the speed of the engine must be considerably reduced to enable the hook from the indicator pulley to be attached to the crosshead.

The only automatic gear for attaching and detaching

the reducing motion from the engine is that now being introduced by Messrs. Crosby and Co. This gear, which we shall describe, enables the operator to attach a complete

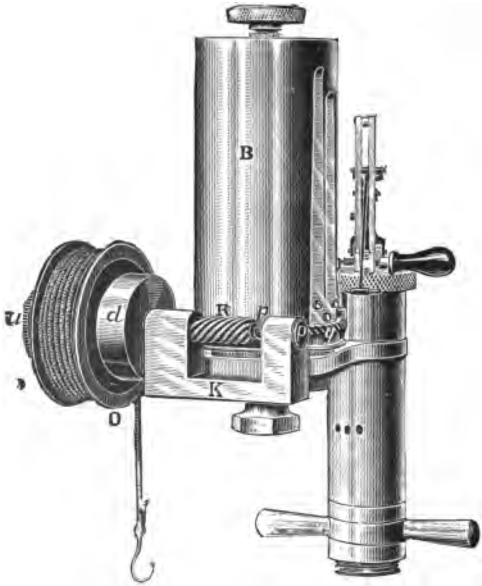


FIG. 77.—Tabor Indicator with Reducing Gear.

indicator gear to any high-speed engine whilst running at full speed Another important feature is that the reducing motion is thrown out of action immediately after the diagram has been taken.

THE WAYNE INDICATOR.

The design of this instrument, which is made by Messrs. Elliott Brothers, is a complete departure from the usual lines. Two views are shown in figs. 78 and 79, from which

the general appearance may be gathered. It will be seen that the paper upon which the diagram is traced is fixed by two spring clips to an aluminium guide, and forms a

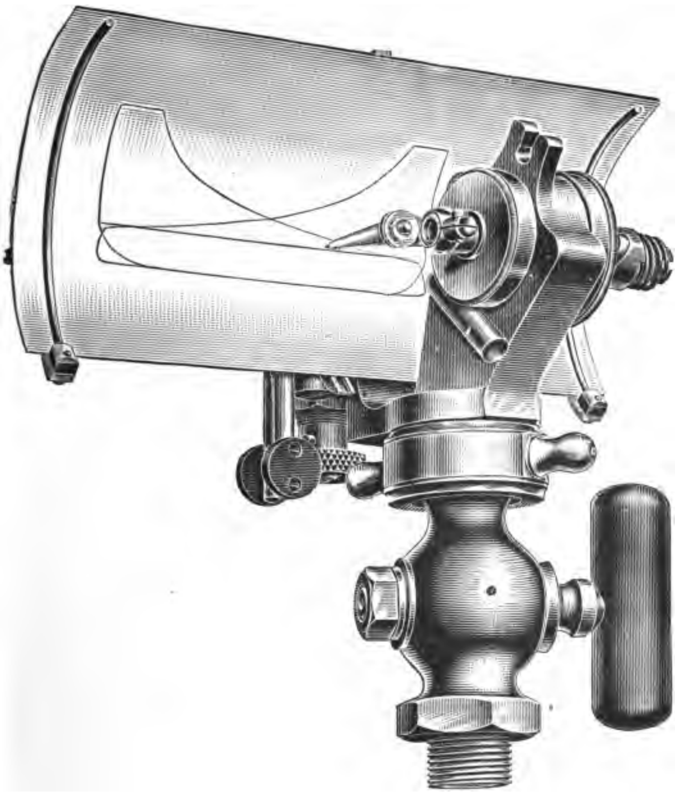


FIG. 78.—Wayne Indicator.

concave surface. The latter moves horizontally by means of a cord passing over a spring pulley and attached to any convenient reducing gear. The marking point traverses

the arc of a circle, its movement being resisted by the torsion of a spiral spring shown on the right of the illustrations. This spring may be changed to suit the

FIG. 79.—Wayne Indicator.

pressures dealt with, and determines the scale of the pressure ordinates. The circular movement of the pencil is produced by the pressure of the steam upon two steel tongues *a*, fig. 80, projecting from a horizontal rod passing

through the cylinder. This rod carries the pencil at one end and the spiral spring at the other. The diagrammatic sketch, fig. 80, will show how the pressure acts upon the steel tongues, and how the steam which passes the tongues escapes through the holes *e*, *e*. The chief features in this indicator are: The absence of joints or parallel motion, small movement of working parts, and accessibility

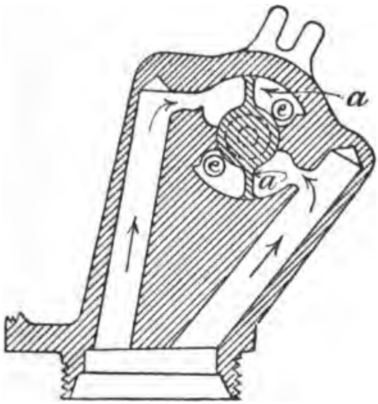


FIG. 80.—Sectional Elevation of the Wayne Indicator.

of the pressure spring. For indicating engines at very high speeds, such as 600 revolutions per minute, an ingenious fitting is added to the instrument, termed by the makers "the lining attachment." By this gear, shown in fig. 79, the indicator diagram is built up line by line, the movement of the pencil for each part added being about one-twentieth of an inch. By turning the handle shown in the front of fig. 79 a worm is rotated. This gears with a segment of a worm wheel, which in turn rotates in a concentric circle with the cylinder. A radial arm, visible in the illustration, fits into a hole in the piston rod, and is attached to the worm-wheel segment by means of a screw passing through a slotted hole. This slotted hole enables

the pencil to move a distance of $\frac{1}{4}$ in. when the worm wheel is stationary. The method of taking a "line" diagram is as follows: First turn the handle operating the worm and worm segment until the slot in the radial arm stands quite free of the set-screw passing through it. This enables the atmospheric line to be accurately taken. Then turn on the indicator cock, and commence to rotate the handle before mentioned. The distance between the horizontal lines of the diagram will depend entirely upon the loss of time occasioned by the slot in the radial arm, and consequently upon the speed of rotation of the handle operating the worm-wheel segment. For any given case a few trials will enable the operator to accommodate the movement of the handle to produce a distance of about one-twentieth of an inch between the horizontal lines, as recommended by the makers. A line diagram is shown in fig. 81. It is

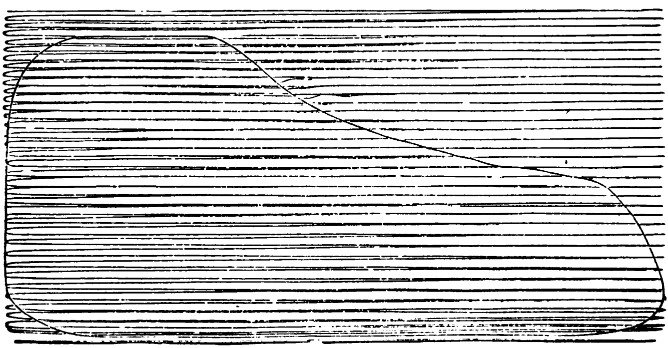


FIG. 81.—Diagram taken with Wayne Indicator.

noticeable that a weaker spring may be used with the line attachment; at the same time a much more accurate diagram is produced. The author has found the lining attachment extremely efficient and simple to work. It may be attached or removed from the instrument in a few

seconds, and certainly produces a most perfect diagram at all speeds. The accessibility of the pressure spring is a great convenience, and its uniform temperature should

FIG. 82.—Simplex Indicator.

increase the degree of accuracy obtained. For convenience in adjusting the instrument it is attached to the cock with a swivel joint.

THE SIMPLEX INDICATOR.

A photograph of this instrument is shown in fig. 82. The instrument is made by Messrs. Elliott Brothers, and is exceedingly strong. The chief departure from the usual construction is in the arrangement and form of spring used to record the pressures. This spring is visible at S. The upper limb is held in the upright standard, which is rigidly attached to the body of the instrument. The lower limb is held in the top of the piston rod of the indicator. When pressure is exerted upon the piston, its movement is resisted by the spring. The spring is quite easily removed by a slight pressure upon its side. Various strengths of springs are supplied.

The pencil levers, which are somewhat heavy, pass through a wide slot cut in the upper part of the instrument, and are attached to the piston rod by a collar, which is free to rotate upon the piston rod. The straight line motion is obtained by four parallel bars, three of which are visible in the photograph.

CHAPTER XI.

Reducing Gear for Gas Engines.

It is invariably necessary to attach to an engine some mechanism to cause the drum of the indicator to reproduce the motion of the piston to a small scale. Such mechanisms are numerous, and for a complete description of the various types the reader is referred to text-books on indicating. The essential requirements of a reducing gear are simplicity, durability, and accuracy. The great fault of nearly all reducing gears arises from the necessity of stopping the engine to attach the gear. This also necessitates the gear running until the engine may be stopped. The wear of the

parts thus becomes abnormal, causing considerable shake and inconvenience, especially in high-speed engines.

Motion for the gear is usually derived from the crosshead of a steam engine or trunk piston of a gas engine. A convenient attachment for the latter is illustrated in fig. 83, at X; the pipe shown is welded to the adjusting screw of the piston pin, and carries the upright bar Y. The lock nuts shown serve to adjust the position of the bar. The motion of Y may be reduced to suit the stroke of the indicator drum in a variety of ways, and a practical

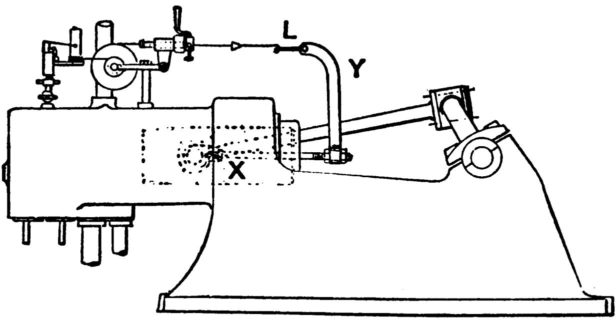


FIG. 83.—General view of Grover's Indicator Gear.

engineer will often be able to devise means suited to individual cases. It is, however, almost essential, to ensure economical working, to have a gas engine tested periodically in order to detect defective valves, and in view of this it is convenient to be able to fit up an entire indicator gear without stopping the engine. In order to secure this condition the author has recently designed and provisionally protected the arrangement illustrated in figs. 83, 84, 85. A number of these gears are being made by Messrs. Crosby and Co., for use in connection with their roll reducing gear and improved indicator.

Referring to fig. 84, it will be seen that the cord from the reducing wheel W passes through a tube, to which a handle

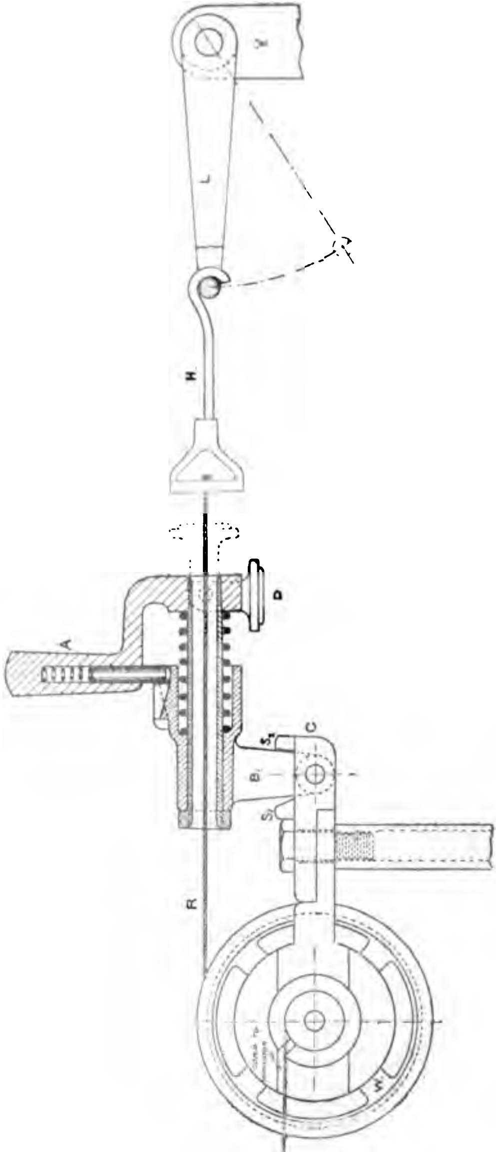


FIG. 54.—Elevation of Grover's Indicator Gear.

is attached. The tube slides in a boss upon the rocking lever B, and is pressed outwards by the spiral spring shown. When the spring is compressed, the bolt fitted in the handle rides up the incline, and, dropping over the edge, prevents the release of the spring. By pushing the handle over, as indicated by the arrow in fig. 85, the spring is released and the bolt in the handle runs up an opposite incline, dropping over its edge, as before. The handle must be pulled back again before the spring can be compressed. Upon the front of the tube a distance piece D is pivoted. This part drops downward by its own weight, but may occupy the horizontal position shown dotted.

A coiled spring inside the reducing wheel W causes the cord R to be in tension; hence the hook H, resting against the front of the handle, tilts the lever B against the stop S_1. When the hook is drawn forward, B rests against the stop S_2.

An important part of the arrangement is the swinging link L attached to the bar Y, figs. 83 and 84. This link, by reason of its inertia, swings from an angle of about 45 deg. into a horizontal position at the end of each instroke. By this means it is made to engage with the hook H when the latter is in position for coupling up.

The only permanent attachment to the engine is the bar Y, with its link L, and in the hands of a skilful operator the whole of the preliminary adjustments of the gear may be made even when running at full speed, though it is advisable to keep a record of the length of hook required for each engine, and to make such observations as will assist in quickly replacing the gear when once adjusted.

We will suppose that our cord from the indicator drum is attached to the reducing wheel, and that all is ready for taking a diagram. The first thing to be done is to compress the spiral spring. If this is omitted the gear may couple up, but it will immediately uncouple itself on the return stroke. Next, draw forwards and *upwards* the hook H, at the same time lifting the distance piece D until it is in the dotted

position, fig. 84. Now allow the hook H to return until it presses against the distance piece D. It will now be found that the distance piece and the hook remain in line, but the hook is tilted upwards because the tension of the cord pulls B against the stop S_1. The gear is now ready for coupling, and this is effected by bringing B gently against the stop S_2. Immediately the hook leaves the distance piece, the latter

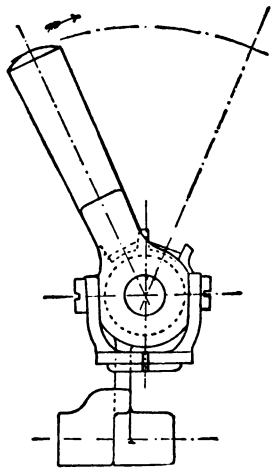


FIG. 85.—End view of parts A, B, and C of Grover's Indicator Gear.

drops down by its own weight, and when next the hook H returns there is a quarter of an inch clearance between it and the face of the tube. The hook therefore remains coupled to the link until the diagram is taken.

To uncouple the gear it is merely necessary to rotate the handle in the direction of the arrow on fig. 85. The spiral spring is then released, the handle flies out, and occupies the position shown in fig. 84. It is obvious that the hook H will now collide with the handle, and its further inward movement will be arrested. Hence the tension of the cord

R will again tend to bring B against the stop S_1. In consequence, the hook H tends to lift from the link L, and when L completes its inward stroke the clearance is so arranged that H is free to tilt upwards, and thus automatically uncouple itself.

CHAPTER XII.

Gas-engine Trials.

When taking indicator diagrams from a gas engine, it is advisable to keep the pencil upon the diagram paper for about half a minute, in order that a series of diagrams may be taken. If the engine is running at full power, the pencil will traverse the same path during each cycle and trace one distinct diagram. If, however, the engine is running lightly loaded, it is possible that it may not receive a full charge of gas, owing to the action of the governor; hence the pencil will trace out a fresh diagram for each cycle. Methods of working out the cards will be given later; at present we shall treat only of the practical details of a test; suffice it to say here that the object of the experimenter should be to obtain a set of diagrams at each operation of the indicator, ranging from a maximum to a minimum area.

The springs used for indicating gas engines are necessarily heavy (from $\frac{1}{120}$ to $\frac{1}{200}$); hence only the compression, explosion, and expansion curves are clearly shown upon the diagram. It is nevertheless important that the small variations in pressure during the exhaust and suction strokes should be ascertained as a check against bad valve setting and undue throttling in the pipes. This is usually done by using a light spring in the indicator (say a $\frac{1}{8}$th). In order to prevent the excessive compression of the spring during the explosion it is usual to pass a $\frac{1}{2}$ in. length of $\frac{3}{16}$ in. or $\frac{1}{4}$ in. brass pipe over the indicator piston rod before

putting the instrument together for use. This acts as a distance piece between the piston and top cover of the indicator cylinder, and thus prevents the spring from being crushed. The length of the brass pipe will depend upon circumstances. In the Crosby indicator a $\frac{5}{16}$ in. length suffices. A diagram taken in this way has been given in fig. 17, and is explained in the text relating thereto. The excessive throttling of a silencing chamber, exhaust pipe, or valves is at once detected by a light spring diagram, and no engine should escape this test after it is completely fitted up.

Measurements of the Jacket Water—In a gas-engine trial there are three observations to be made with respect to the jacket water :—

1. The quantity (measured in pounds).
2. The temperature of the inlet (measured in deg. Fah.).
3. The temperature of the outlet (measured in deg. Fah).

The Quantity.—The most convenient method of measuring the weight of jacket water depends so much upon local conditions that no specific advice may be given. The author has found the following method convenient, and of wide application: Most engines not specially fitted for testing will be arranged as in fig. 2. This being so, empty the tank B by syphon or otherwise, and plug the orifice of the upper circulating pipe leading into tank A. No circulation through the jacket is now possible without opening the feed cock to A. The feed water delivered to A will pass through the jacket and find its way into tank B. This tank may be calibrated to read pounds of water, either by means of a float or by the insertion of a glass tube passing up the outside of the tank. In some instances it has been less trouble to uncouple a union on the pipe *e* and conduct the water to a tank resting upon a weighing machine. In either case the quantity should be regulated by the feed pipe to A, until the temperature of the outlet water is about 150 deg. Fah. Observations of the weight of water may be taken every ten minutes, though with a steady flow

this is not necessary. When short handed only the total weight at the end of the trial need be observed.

Jacket Temperatures, Inlet.—The inlet temperatures should be taken on the inlet pipe near the tank A. If convenient, insert a ⊥ piece in the inlet pipe, and through a cork in the ⊥ place a Fahrenheit thermometer vertically, so that the mercury bulb is directly in the stream of water. Many errors arise through inattention to this. If the vertical part of the ⊥ is long, a small volume of stagnant water remains in it and gathers heat from the engine. The thermometer readings may be affected by this, and become altogether unreliable. In most cases the inlet temperatures may be observed from the water in the tank A, thus obviating any alteration to the pipes.

The Outlet Temperature.—This should be taken as near the exit from the jacket as possible. A convenient place is in the bend at the top of the pipe *e*. The thermometer should be placed vertically, especially if permanently fitted, as, if not, the column of mercury tends to stick at the maximum temperature recorded. It has been stated above that the usual outlet temperature is about 150 deg. Fah. The thermal efficiency of an engine may be increased by raising the temperature of the jacket. There is therefore a tendency to raise the temperature of the jacket beyond its safe working limit when running short economy trials. An increase in efficiency acquired in this way is sometimes ignorantly ascribed to other causes, and fair comparisons cannot be made without regard to the temperature of the jacket outlet water. No doubt a time will come when the jacket, as now usually fitted, will be dispensed with; for, although the gas engine is—theoretically and practically—the most efficient heat engine, it is nevertheless a humiliating admission that provision must be made for wasting between 30 and 40 per cent of the heat generated. Where hot water is required in factories there is no reason why this heat should not be utilised by a convenient arrangement of service pipes. The author has arranged a plant

to work upon this system, which will effect a considerable saving in fuel, and be otherwise extremely convenient.

Measurement of the Gas.—The number of cubic feet of gas used, also the temperature and pressure of the gas in the meter, require to be known. The index of an ordinary gas meter is usually provided with a set of three or four pointers indicating respectively thousands, tens of thousands, hundreds of thousands, &c., of cubic feet of gas. Besides these, there is a smaller pointer placed above, which revolves once for, say, 10, 20, or 30 cubic feet according to the size of the meter. Besides these, another figure, which is frequently misleading, will be found printed upon the dial. This figure refers to the capacity of the measuring chamber inside the meter, and this is frequently noted upon the dial in the following way: "1·32 cub. ft. per rev." Mistakes occur in supposing this figure, 1·32 cubic feet, to refer to a revolution of the small pointer, instead of, as above mentioned, to the capacity of the measuring chamber. It should, therefore, always be remembered that this figure has absolutely no reference to the pointers on the dial.

When the meter is situated in a room of practically uniform temperature, the meter may be assumed to be of the same temperature as the air in the room. It is, however, more satisfactory to have a thermometer fitted to the exit pipe from the meter, so that the temperature of the gas may be more accurately ascertained. The pressure of the gas is best ascertained by the application of a manometer, or U tube filled with water, to some connection close to the meter. The head of water sustained by the pressure of gas should be noted a few times during the trial, to see that no variation of pressure takes place. In connection with the observation it is also necessary to take the height of the barometer, so that all readings may afterwards be reduced to one standard of atmospheric pressure.

Junker's Calorimeter.—In very complete trials it is necessary to know the average composition of the gas during the trial, though in most tests the calorific value of the gas

(which, for general purposes, is the only figure required) may be ascertained by means of an instrument known as Junker's calorimeter. This instrument, which may be obtained from the sole agent, Hermon Kühne, New Broad Street, E.C., is illustrated in figs. 86 and 87. The principle of the instrument is that the heat generated by a gas flame is absorbed by a water jacket. The quantity of water, its inlet and outlet temperatures, and the quantity of gas passing through the instrument, afford data from which the calorific value of the gas may be determined to within 0·5 per cent of any other determination. Radiation is prevented by surrounding the apparatus by an air jacket formed by a nickel-plated cylinder. The flame shown at 28, fig. 86, is surrounded by a water jacket, through which pass a number of vertical copper tubes. The flame burns in the central chamber, and the products of combustion pass down the inside of the tubes to the outlet at 33. By this arrangement the gases at the highest temperature meet the hottest water, whilst as the gases cool they meet the colder inlet water. This, of course, favours the transmission of heat, and the products of combustion escape at the throttle at atmospheric temperature, having parted with the heat of combustion to the jacket water. In order to obtain accurate results, it is necessary that the flow of water through the instrument should be perfectly uniform. This is secured by leading the water into a tank shown at 3, fig. 86. This tank is kept slightly overflowing, thus producing a constant head. Two thermometers are fitted for measuring the temperatures of the inlet and outlet, and a graduated measuring glass is supplied for water measurements. It is highly important that the gas pressure remain constant, and for this reason a suitable regulator is required to be used with the calorimeter. An external view of the apparatus showing the gas meter and regulator is given in fig. 87. Water formed by the combustion in the central chamber is collected in a small measuring glass at *d*, fig. 87.

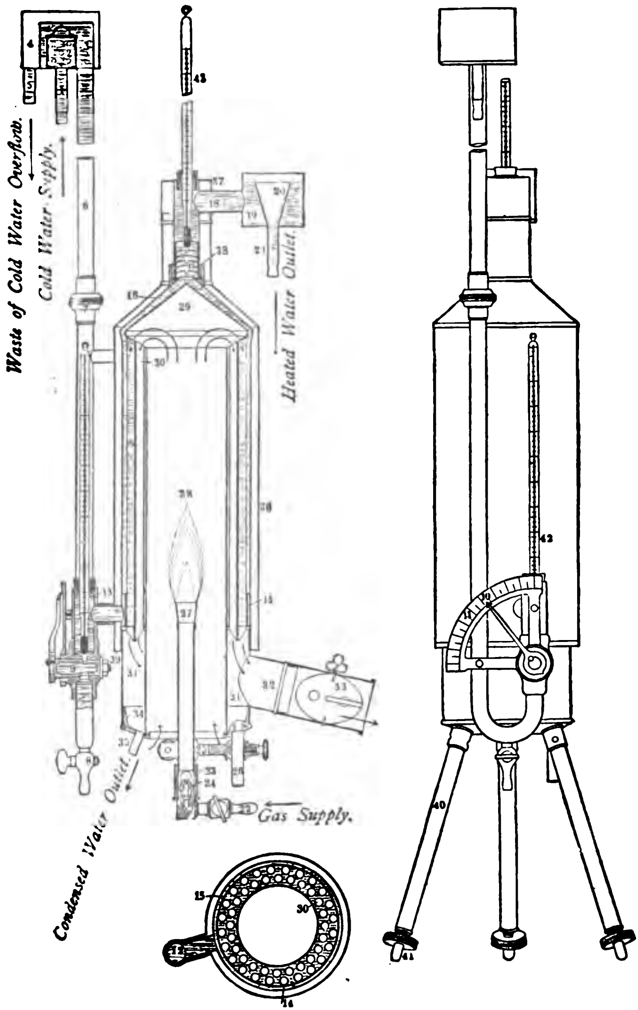


FIG. 86.—Section and Elevation of Junker's Calorimeter.

The calibrations of the instrument have been made in the metric system, and the unit of heat has been taken as the quantity required to raise the temperature of 1 kilogramme of water 1 deg. Cen. This unit is termed a calorie, and is converted into British thermal units* by multiplication by 3·96.

In fixing the apparatus, the chief points to be observed are as follow :—

No draughts must be allowed to strike the outlet of the exhaust gases. The quantity of gas passed through the calorimeter should be at the rate of

4 to 8	cubic feet	per hour of	illuminating gas,
8 to 16	„	„	hydrogen,
16 to 32	„	„	Dowson gas.

Always light the burner outside the combustion chamber, to avoid explosion due to accumulated gas.

Regulate the water passing through the jackets until its rise of temperature is from 10 to 20 deg. Cen.

Having set up the apparatus and regulated the quantity of gas and water, a test may be made in the following way: Observe the reading of the gas meter at a convenient figure, and at the same time remove the tube C, fig. 87, to discharge into the graduated glass. Then take the readings of the inlet and outlet thermometers. Take a few intermediate readings of the outlet thermometer, and immediately the water reaches, say, the 2 litre mark, turn off the gas and read the quantity of gas shown by the meter. The readings may be booked as follow :—

Gas meter.	Cold water thermometer.	Hot water thermometer.	Water jacket.
4 cubic feet ...	7·98 ...	27·54 ...	0
	„ ...	27·52 ...	
	„ ...	27·53 ...	
4 268 cubic feet	„ ...	27·55 ...	2 litres
Gas burnt 0·268 cubic feet	7·98 ...	27·53 (mean)	2 litres

* The heat required to raise 1 lb. of water of maximum density 1 deg. Fah. is equal to one British thermal unit.

Mean rise in temperature

$$= 27{\cdot}53 - 7{\cdot}98 = 19{\cdot}55.$$

Calorific value of gas in calories per cubic foot

$$= \frac{\text{mean rise in temperature} \times \text{litres of water}}{\text{cubic feet gas burnt}}$$

$$= \frac{19{\cdot}55 \times 2}{0{\cdot}268} = 145{\cdot}8 \text{ calories}$$

$$= 145{\cdot}5 \times 3{\cdot}96 = 577 \text{ B.T.U.}$$

So far we have obtained a calorific value of the gas without regard to any condensed water which may have collected

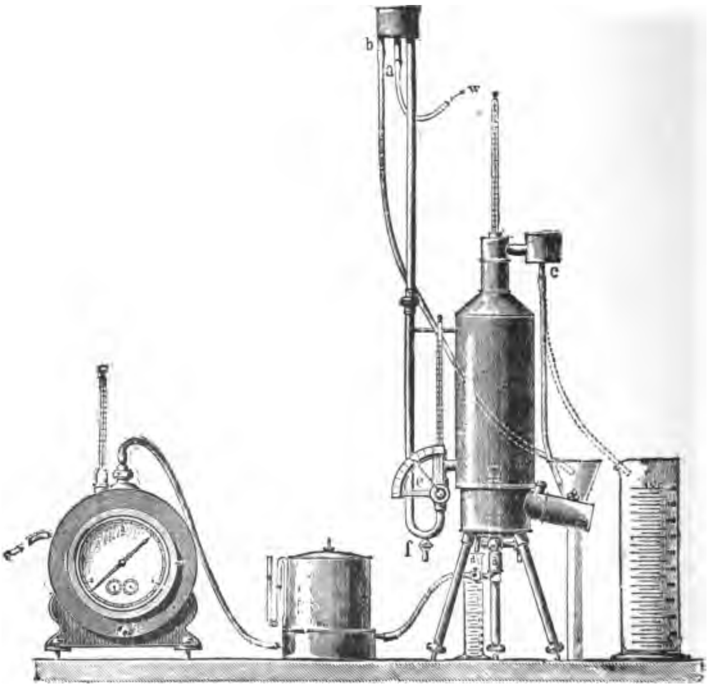


FIG. 87.—External View of Junker's Calorimeter.

in the small measuring glass marked D, in fig. 87. Now, this moisture is the result of the combustion of hydrogen with the oxygen of the air, and at the moment of combustion steam is formed. In the calorimeter this steam is condensed, and gives up its latent heat to the calorimeter; hence the calorific value above obtained is the total heat evolved per pound of gas when all the products of combustion are cooled to atmospheric temperature.

In nearly all industrial processes the steam formed by the combustion of hydrogen with oxygen passes away as steam; consequently some of the heat is not available unless the steam is all condensed to water. The number of British thermal units carried away per pound of hydrogen is, in round numbers, 10,000. This is obtained in the following way: 1 lb. of hydrogen requires eight times its weight of oxygen to completely burn it; hence 9 lb. of steam would be produced. Suppose this to be cooled to water at 60 deg. Fah., and at atmospheric pressure, the heat evolved by the process would be equal to $9 \times \{966 + (212 - 60)\}$ units per pound of hydrogen, which amounts to 10,062 British thermal units. It is evident, therefore, that two calorific values may be obtained—the one the *total*, the other the *available*. In gas engines the heat of the exhaust does not permit of condensation of the steam produced; hence the latent heat should be subtracted to give the available heat for use in the gas engine.

For this determination it is advisable to continue the operation of the calorimeter for a longer time, because of the small quantity of condensed water produced. Suppose the measuring glass D contains 60 cubic centimeters after burning 3 cubic feet of gas; then the correction in calories

$$= \frac{\text{cubic centimeters of water} \times 0{\cdot}6}{\text{cubic feet gas burnt}}$$

$$= \frac{60 \times 0{\cdot}6}{3}$$

$$= 12 \text{ calories.}$$

Then the net available heat

$$= 145.8 - 12 = 133.8 \text{ calories}$$
$$= 529 \text{ B.T.U.}$$

It is sometimes argued that in estimating the efficiency of a gas engine the total heat, and not the net available heat, should be taken as the basis. Nearly all efficiency trials have hitherto been worked upon the available calorific value of the gas used, and for the sake of uniformity we shall adopt this standard. After all, the question is merely one of definition, and needs no further discussion here.

Method of Taking Sample of Gas.—Should it be desired to take a continuous sample of gas for future analysis, the following arrangement will be found convenient: Fit an indiarubber cork into a suitable glass jar, and through two holes insert glass tubes. The one tube should be short, and the other should reach to the bottom of the jar. Attach the shorter tube to the gas supply by means of an india-rubber pipe, and form a syphon of the other longer tube by another piece of rubber pipe. Having completely filled the apparatus with water, a pinch cock may be adjusted on the syphon so that the latter draws the water slowly from the bottle or jar. Gas then enters the jar, and by suitably regulating the syphon small quantities of gas may continuously enter the jar during the whole period of the trial. In this way a fair representative sample of gas is obtained for analysis. Mercury is, of course, preferable to water, inasmuch as the constituents of coal gas are more or less soluble in water. No serious errors arise provided the gas is not left for a long period in contact with a large surface of water. Precise instructions for analysing gases will be given in another chapter.

Counting the Revolutions and Explosions.—The revolutions of the crank shaft are usually observed by means of some form of speed counter. The following methods of actuating the counter have been made use of by the author, and are given here as suggestions. It often happens that

incidental conditions adapt themselves to the requirements of the experiments, and for this reason it is always well to be on the alert rather than to attempt to laboriously carry out a special method of fixing the measuring apparatus.

Suppose the counter to have a reciprocating lever which requires to be moved through two short strokes at each revolution of the crank shaft. When the reducing gear for indicating is constantly working throughout the trial, motion may be transmitted from some of its bars by a cord attached to the lever of the counter. The cord may be held in tension by attaching a strong indiarubber band to the counter lever. Indiarubber bands are so extremely useful in fixing up testing apparatus that they should be regarded as part of the equipment for an engine trial. Should the reducing gear be unsuitable for driving the counter, motion may be derived from the valve levers. In this case it must be borne in mind that the side shaft of an Otto cycle engine is geared down to run at one-half the revolutions of the crank shaft; hence the counter readings must be doubled. For the *permanent* attachment of a counter it is, of course, better to employ a link or rod for operating the counter lever, though the cord and indiarubber band is much more easily fitted up, and is just as reliable for a three or four hours' test. Small counters are procurable which may be held against the centre of the crank shaft by hand. Some are arranged to give the revolutions per minute on the dial, others give the revolutions during the time the counter is held in position.

The explosions are best counted from the exhaust pipe outlet. If the engine is running at full power, and is working on the Otto cycle, the explosions will, of course, be equal to half the revolutions. The explosions should in any case be counted occasionally, in order to verify the assumption, as derangement of the governor gear and ignition tube may falsify the results if not checked.

Temperature of the Exhaust.—This measurement is difficult to obtain with any approach to accuracy. It is indeed

often left to be calculated from the indicator card. This, however, is not satisfactory, because of the alteration of the specific heat of the gases forming the products of combustion. It has been shown by Le Chatelier that the specific heat of carbon-dioxide increases in the ratio of 1 to 1·6 between temperatures of 0 and 1,000 Cen. Until more is known of the specific heats of gases at high temperatures, calculations from the cards can only be regarded as approximating to the results required.

Mr. Burstall has carried out experiments in this direction, using a fine platinum wire suspended in the combustion chamber. The electrical resistance was measured, and from this the temperature deduced. This was found to range from 1,045 deg. Cen. to 1,140 deg. Cen. At the present time there is no simple and reliable instrument which can readily be fitted up for measuring directly the exhaust temperatures; hence most experimenters have to adopt a method of calculation from the indicator diagram.

We may now summarise the observations during a gas-engine trial as follows:—

Time of observations.

Brake readings $\begin{cases} \text{Spring balance.} \\ \text{Load on brake.} \end{cases}$

Revolutions of the engine by counter.

Explosions per minute.

Gas meter $\begin{cases} \text{Quantity, cubic feet.} \\ \text{Temperature.} \\ \text{Pressure.} \end{cases}$

Jacket water $\begin{cases} \text{Quantity, pounds.} \\ \text{Inlet temperature.} \\ \text{Outlet temperature.} \end{cases}$

Height of barometer.

Indicator diagrams.

Temperature of exhaust (when possible).

CHAPTER XIII.

THE PRACTICAL ANALYSIS OF COAL GAS.

Analysing the Gas.—Hempel's apparatus is most generally used by engineers for the analysis of furnace and coal gases. If carefully handled, the results may be relied upon as being accurate to the half of 1 per cent. The principle upon which the analysis depends is briefly as follows: From a known volume of gas certain constituents are absorbed. The remaining volume is measured after each absorption is complete; hence the volumetric proportion of the constituents may be determined. If there is a residue which cannot be absorbed, as is the case with coal gas, the remaining quantities may be determined by combustion in a manner to be described.

We will suppose that a sample of coal gas has been collected during the trial, as previously described. This sample, we know, consists of carbon-dioxide (CO_2), olefines (C_6H_4), oxygen (O), carbon-monoxide (CO), hydrogen (H), and marsh gas (CH_4). Our analysis will enable us to determine the volumetric proportions of these constituents, but it will be noticed that the method depends entirely upon our knowledge of their presence. The constituents must be absorbed in the following order:—

1. Carbon-dioxide, absorbed by potassium hydrate.
2. Olefines, aborbed by strong sulphuric acid.
3. Oxygen, absorbed by phosphorus or pyrogallic acid.
4. Carbon-monoxide, absorbed by cuprous chloride dissolved in hydrochloric acid.

The apparatus for carrying out these processes is illustrated in figs. 88 and 89. A is a plain tube, called the levelling tube; B is a graduated tube, called the burette. The burette is graduated up to 100 cubic centimetres. The two tubes are connected together by means of a flexible rubber pipe, which may be of comparatively thin section

when water is used in the apparatus, but should be much stouter tubing when mercury is used.

There is a cock at the top of the buretted and a pinch cock upon the rubber connection. The level tube is first

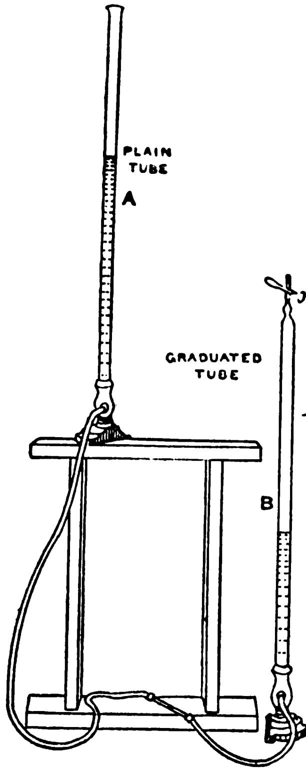


FIG. 88.—Hempel's gas burettes.

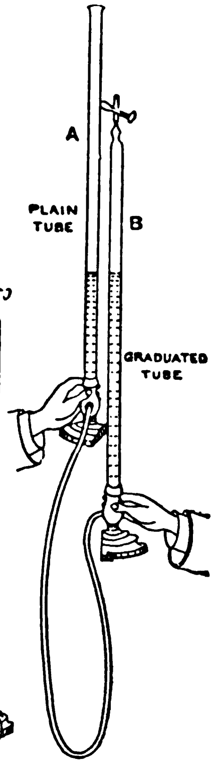


FIG. 89.—Method of levelling when measuring volume of gas.

filled with water (or mercury if procurable). The burette is then lowered until the water gravitates into it and

completely fills it up to the stop cock. Now connect the syphon tube of the sample bottle to a water tap, and gently turn on the water so as slightly to raise the pressure of the sample gas. Allow a little gas to escape through the other tube from the sample bottle, in order to get rid of any air beyond the pinch cock. The sample may now be connected to the burette, and the water run from the burette back into the level tube. The sample gas will then be drawn into the burette. It is better to draw in rather more than

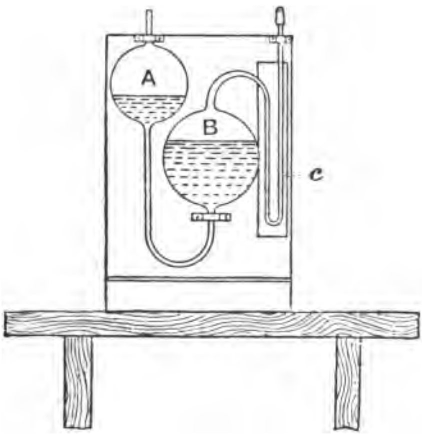


FIG. 90.—Hempel's simple absorption pipette.

100 c.c. before disconnecting the sample bottle. This done, close the stop cock of the burette, and raise the level tube until the surface of the water stands at zero. Close the pinch cock on the flexible tube. The pressure in the burette will now be slightly above that of the atmosphere; hence the stop cock of the burette should be opened momentarily just to allow the excess of pressure to escape. This method of filling contributes towards accuracy, for it is of the highest importance that no measurements of volumes be taken excepting when the gas is at atmospheric pressure. It is

also of the highest importance that the temperature of the burette remain constant, and for this reason the tubes should never be touched; all lifting should be done by holding the wooden stands into which the tubes are fitted. By drawing the hand once or twice down the burette, an alteration of quite 5 per cent will be noticed in the reading, thus showing the importance of uniformity of temperature throughout the analysis.

Having measured off our 100 c.c. at atmospheric pressure, we next proceed to absorb the carbon-dioxide. Each absorption is carried out in a pipette, the forms of which are shown in figs. 90, 91, and 92. Fig. 92 represents two glass bulbs connected together by glass tubes as shown.

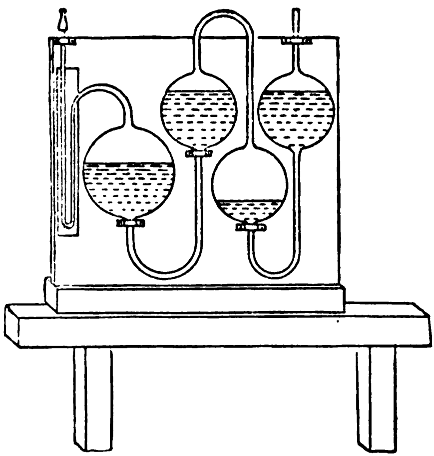


FIG. 91.—Hempel's double absorption pipette.

The larger bulb is filled with a strong solution of potassium hydrate until the capillary U tube fills. It is advantageous to place a number of $\frac{1}{8}$ in. solid glass rods inside the long bulb. The solution adheres to the surface of the rods, and thus presents a very large surface for absorbing the CO_2.

After making sure that the solution is well up the capillary tube, couple the burette to the pipette, as in fig. 93, by a very short piece of capillary tube. The admission of air in coupling and uncoupling with the various pipettes used is a frequent source of error, and the utmost care is required. A con-

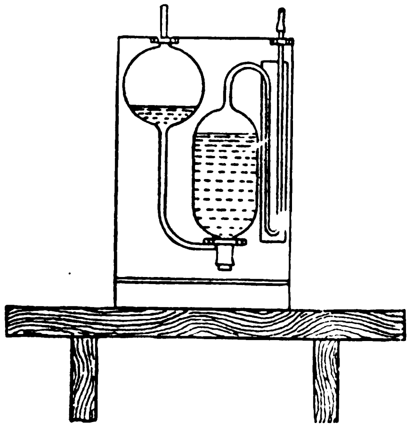


FIG. 92.—Hempel's pipette for solid substances.

venient way of making these joints is to slip a $\frac{3}{4}$ in. length of rubber tube over and beyond the end of one tube. Then butt the two ends together, and by wetting the glass the rubber can be slipped over the joint without enclosing a large volume of air. When the joint is made, open the stop cock, raise the level tube, and drive all the gas over into the pipette. The absorption will be complete in about half an hour. The process will be assisted by shaking the solution or by drawing the gas back into the burette, so that the glass rods may be again moistened.

When the process of absorption is completed, lower the level tube and allow the water to flow from the burette until the caustic potash in the pipette rises up the capillary tube to the position it at first occupied. Now close the stop

cock at the top of the burette, and hold the level tube against the burette at such a height that the water in both tubes is exactly at the same level. In this way the gases in the burette are always brought to atmospheric pressure before their volume is measured. In taking the readings great

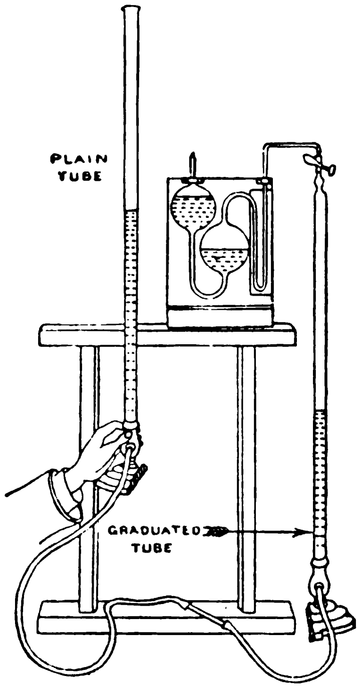


FIG. 93.—Method of using Hempel's apparatus.

care should be exercised. When water is used its surface in the tube will be concave; with mercury the surface is convex. In either case the line of sight when taking a reading should be horizontal and tangential at the *centre* of the concave or convex surfaces. Before taking a reading

when water is used in the tubes, the burette should be allowed to stand a few moments to allow drops to run down the sides. It is only by careful attention to such details that any approach to accuracy can be secured. An exactly similar process is carried out in order to absorb the olefines. The pipette used is shown in fig. 90, and contains strong sulphuric acid.

To absorb the oxygen, a pipette similar to fig. 92 is used. This should be filled with water, and contain a number of sticks of phosphorus of about ⅛ in. diameter. As phosphorus is not readily procured in such thin sticks, it will be well to describe how they may easily be produced from lumps of phosphorus without danger to the operator. Procure a glass tube, the bore of which is equal to the diameter of the sticks required. At the end of this place a short length of rubber pipe. Take a tumbler of water, at about 120 deg. Fah., and quickly drop the lumps of phosphorus into the hot water. The phosphorus will now melt at the bottom of the tumbler. Squeeze the rubber pipe attached to the glass tube, and then immerse the end of the glass tube in the melted phosphorus. Owing to the head of water in the glass, it will be found that the phosphorus will run up the glass tube when the rubber at the other end is released. By suitably arranging the head of water, sticks of phosphorus of the required length may be cast. To get the stick of phosphorus from the glass tube it is advisable to remove the rod into a vessel of cold water, but before doing so take care to pinch tightly the rubber connection, so that the phosphorus does not drop when taken from the water. In a few seconds the phosphorus will set, and will stick in the tube if not prevented. Whilst setting the phosphorus should be kept moving in the glass tube by pressing the rubber tube in one or two places in addition to keeping its end tightly closed. A little practice will enable the operator to cast phosphorus safely and quickly. It is extremely important in dealing with such an inflammable material as phosphorus that it

should be handled always under cold water, and never be left exposed to the air for many seconds.

The absorption of oxygen is carried out exactly as previously described.

The carbon-monoxide (CO) is absorbed in a double pipette, as shown in fig. 91. The two bulbs on the right contain water, whilst the two bulbs on the left attached to the capillary tube contain the cuprous chloride and hydrochloric acid. In manipulating this pipette great care should be taken not to pass the cuprous chloride over into the water bulb. If this be done, a white precipitate of cuprous chloride will be formed in the water chamber, and if left may block up the passage between the water bulbs. With this additional caution, the work may be continued as previously described.

We have now absorbed four of the constituents of the gas, and we have a residue of hydrogen and marsh gas. To evaluate these quantities an entirely different method is employed.

To effect the explosion of the hydrogen and marsh gases, a pipette of the form shown in fig. 90 is used, having two platinum wires fused into the bulb B, terminating in sparking points on the inside of the bulb. See that the bulb B and the capillary tube C are filled with water. Then drive from the burette 10 c.c. of the residue into the pipette. Then, either by emptying the burette, or, preferably, by using a second burette, drive seven times the volume of air—that is, 70 c.c. of pure air—into the pipette, as a supply of oxygen for the explosion. It will be found that this charge occupies about half the volume of B. This should not be exceeded. By means of a bichromate battery and induction coil send a spark through the explosive gas, and violent combustion will follow. On account of the sudden pressure, it is well to bind the rubber connection to the pipette with wire. Sometimes a glass stop cock is provided between the chambers A and B. If so, this cock must be *open* when the explosion

takes place, or the bulb B will be broken by the pressure. After the explosion measure the volume of the remaining gas by drawing it back into the burette, observing the cautions already given. It will be found that there is a considerable diminution in volume, due to the free hydrogen and the hydrogen in the marsh gas combining with oxygen of the air to form steam, which ultimately is condensed on the sides of the pipette to a negligible volume of water. The gases remaining after the explosion consist of carbon-dioxide—the result of the combustion—nitrogen, which was drawn in with the air, and an excess of oxygen above that required for the complete combustion of the hydrogen and marsh gas. It will be observed that the diminution in volume after the explosion is due to the disappearance of gaseous hydrogen from two constituents, namely, the free hydrogen and that contained in the marsh gas. It is known that one volume of marsh gas when completely burnt with oxygen forms an equal volume of carbon-dioxide. If, therefore, we absorb the carbon-dioxide by means of the caustic potash pipette, we shall obtain the volume of the CO_2 formed by the explosion. This volume gives the required volume of marsh gas. The remainder of the diminution of volume after the explosion must therefore be due to combination of free hydrogen with oxygen. And we know that one volume of oxygen combines with two volumes of hydrogen; hence of the three volumes disappearing due to hydrogen two-thirds of this contraction gives the volume of hydrogen. The nitrogen in the coal gas may be estimated "by difference." But it is preferable to absorb all the oxygen from the residue, and so obtain a residue of nitrogen only. The nitrogen contained in the air used for the explosion may be calculated, and subtracting this from the total remainder of nitrogen gives the volume due to the coal gas.

As an example of the process, the following analysis of Leeds coal gas, obtained by the author in 1895, is fully worked out:—

90 c.c. of coal gas were taken into the burette.

1. Volume of water in burette before any constituents were absorbed was ... 10 c.c.
   Volume of water in burette after absorption of carbon-dioxide (CO_2) ... 10·1 c.c.
   Therefore volume of CO_2 in 90 c.c. of coal gas ... = 0·1 c.c.
   Equivalent volume of CO_2 in 100 c.c. of gas ... = 0·11 c.c.
2. Volume of water in burette before absorption of olefines (C_6H_4) ... = 10·1 c.c.
   Volume of water in burette after absorption of olefines (C_6H_4) ... = 12·6 c.c.
   Therefore volume of C_6H_4 in 90 c.c. of gas ... = 2·5 c.c.
   Equivalent volume of C_6H_4 in 100 c.c. of gas ... = 2·77 c.c.
3. No oxygen was detected.
4. Volume of water in burette before absorption of carbon-monoxide (CO) ... = 12·6 c.c.
   Volume of water in burette after absorption of carbon-monoxide (CO) ... = 20·0 c.c.
   Therefore volume of CO in 90 c.c. of coal gas ... = 7·4 c.c.
   Equivalent volume of CO in 100 c.c. of gas ... = 8·22 c.c.

10 c.c. of residue were taken into burette, and mixed with 80 c.c. of air; hence—

Volume of water in burette before explosion ... = 10 c.c.
Volume of water in burette after explosion ... = 26 c.c.
Therefore total contraction in volume was ... 16 c.c.
Volume of water in burette before absorbing CO_2 formed by explosion ... = 26 c.c.
Volume of water in burette after absorbing CO_2 formed by explosion ... = 30·7 c.c.
Therefore CO_2 = 4·7—that is, volume of CH_4 in 10 c.c. of residue ... = 4·7 c.c.

But if 100 c.c. of coal gas had been taken into burette, then the total residue of CH_4 and hydrogen would have been—

$$\begin{aligned} & 100 - (0{\cdot}11 + 2{\cdot}77 + 8{\cdot}22) \\ = \; & 100 - 11{\cdot}10 \\ = \; & 88{\cdot}9 \text{ residue of } CH_4 \text{ and H.} \end{aligned}$$

Therefore the total CH_4 will be

$$4{\cdot}7 \times \frac{88{\cdot}9}{10} = 41{\cdot}78 \text{ c.c.}$$

Now, we have seen that one volume of CH_4 combines with two volumes of O_2 to form one volume of CO_2 and two volumes of H_2O. The two volumes of H_2O condense to water, and are quite negligible. Therefore we see that the

diminution in volume due to CH_4 in the exploded gases is equal to twice the volume of CO_2 formed by the combustion.

Volume of CO_2 absorbed from products of combustion $= 4{\cdot}7$ c.c.
Total contraction of volume after explosion $= 16$ c.c.
Therefore contraction due to free hydrogen
$=$ total contraction $-$ contraction due to CH_4
$= 16 - 4{\cdot}7 \times 2 = 16 - 9{\cdot}4 = 6{\cdot}6$ c.c.

Now, two volumes of hydrogen combine with one volume of oxygen, and all three volumes disappear after the steam formed by combustion is condensed. It is evident, therefore, that out of the 6·6 c.c. of contracted volume only two-thirds was free hydrogen.

The volume of hydrogen in 10 c.c. of residue

$$= 6{\cdot}6 \times \tfrac{2}{3} = 4{\cdot}4.$$

Therefore the volume of hydrogen in total residue, namely,

$$88{\cdot}9 \text{ c.c.} = 4{\cdot}4 \times \frac{88{\cdot}9}{10} = 39{\cdot}11.$$

Out of 10 c.c. of residue we have obtained 4·4 hydrogen and 4·7 CH_4, which together make up 9·1 c.c.; the remaining 0·9 c.c. should be the volume of nitrogen in the coal gas. Therefore, by difference, the nitrogen amounts to

$$0{\cdot}9 \times \frac{88{\cdot}9}{10} = 8{\cdot}0 \text{ c.c. (nearly).}$$

But our analysis ought to be further checked, and we will proceed to estimate the nitrogen by direct measurement.

Our residue now consists of nitrogen due to coal gas + nitrogen due to air taken in for the explosion + excess of oxygen in the air not required to complete the combustion.

If we absorb the excess of oxygen, we shall at once obtain the whole volume of nitrogen present.

Referring to last reading of burette, we find—

Volume of water in burette before absorbing oxygen, 30·7 c.c.

Volume of water in burette after absorbing oxygen, 35·9 c.c.

Therefore excess of oxygen = 5·2 c.c.

Therefore the total nitrogen left in burette = 100 − 35·9 = 64·1 c.c.

The percentage of nitrogen in air is approximately 79 per cent; therefore out of 80 c.c. taken into burette the nitrogen

$$= 80 \times \frac{79}{100} = 63{\cdot}2 \text{ c.c.}$$

Therefore the volume of nitrogen due to the coal gas

$$= 64{\cdot}1 - 63{\cdot}2 = 0{\cdot}9 \text{ c.c. (nearly).}$$

This agrees with our determination by difference; hence we know our analysis to be very approximately correct.

Collecting our results, we get—

Volume per cent of	CO_2	= 0·11	
,, ,,	C_6H_4	= 2·77	
,, ,,	O	= 0·00	
,, ,,	CO	= 8·22	
,, ,,	H	= 39·11	
,, ,,	CH_4	= 41·78	
,, ,,	N	= 8·00	(by direct measurement)
		99·99	error 0·01.*

The various producer gases may be analysed by Hempel's apparatus in the manner described. The following tables give the composition of coal and producer gases analysed in

* The author regards this near approximation to accuracy as the result of rather larger *compensating* errors in the work. Nothing less than 0·1 c.c. can be read with certainty upon the burette scale; hence an error of 0·01 on the analysis may give a false impression of the accuracy obtainable with this apparatus.

connection with gas-engine trials. It will be understood that the figures given below refer to particular samples of town gases, and that the daily variation in the composition always necessitates an analysis when the constituents of the gas are required with accuracy.

Composition of Coal Gas in Various Cities.

Percentage by volume. Constituents in gas.	Leeds, 1895 (Grover).	London Society of Arts trial.	Kilmarnock (Professor Kennedy).	Paris (Witz).
Marsh gas CH_4	41·78	37·34	42 80	32·30
Olefines C_2H_4................	2·77	3 77	5·55	5·50
Hydrogen H	39 11	50·44	43·60	52·80
Carbon-monoxide	8·22	3·96	4·30	5·60
Nitrogen	8·00	3·98	2·70	3·80
Carbon-dioxide and oxygen .. (Chiefly CO_2 with traces of O.)	0·12	0·51	5·35	0·00

Composition of Poor Gases.*

Name of gas.	Oxygen. Vol. %	Hydrogen. Vol. %	Marsh gas. Vol. %	Olefiant gas. Vol. %	Carbon-monoxide. Vol. %	Carbon-dioxide. Vol. %	Nitrogen. Vol. %
Siemens producer gas	..	8 60	2·40	..	24·40	5·20	59·40
Water gas	0·10	50·50	0·60	..	44·40	1·60	..
Strong gas	..	53·00	..	..	35·00	4·00	8·00
Lowe gas	..	30·00	..	..	28·00	34·00	8·00
Dowson gas	0·03	18·73	0·31	0·31	25·07	6·57	48·98
Dowson gas	0·23	24·36	1·16	0·15	17·55	6·07	50·48
Lenchanchez	0·50	20·00	..	4·0	21·00	5·00	49·50

* From "Donkin on Gas, Oil, and Air Engines."

Orsat Gas Sampling Apparatus.

A portable and convenient arrangement for gas analysis is that known as the Orsat apparatus. This is illustrated in Fig. 94, and consists of a water-jacketed burette permanently attached to the pipettes, usually four in number. The apparatus is designed more particularly for the analysis of furnace gases, but coal gas may readily be analysed by using in addition to the pipettes permanently attached such others in the Hempel form as may be necessary.

The pipettes A, B, C, contain respectively potassium hydrate, phosphorus, and cuprous chloride, the latter dissolved in hydrochloric acid. The pipette C_1 contains water, and acts merely as a reservoir. Producer gases yield only traces of olefiant gas, and except in very special cases this may be neglected. The constituents to be determined in producer gases are Hydrogen, occasionally Marsh gas, Carbonic oxide and Carbonic acid. For the determination of such constituents, the apparatus as shown in the drawing is all that is necessary, and we may proceed to describe its manipulation in testing a sample of Dowson gas.

Before attempting to use the apparatus all the glass plugs should be withdrawn and covered with a thin coating of vaselene. If the plugs should be stuck, they may be loosened by applying hot wet cloths to them, at the same time putting a little methylated spirit on the plug, so that as much as possible will find its way between the adhering surfaces. When the plugs are withdrawn for greasing, the liquid in the pipettes will fall to a lower level. When the stoppers are replaced the liquid in each pipette must be drawn up into the capillary tube in which the pipette terminates. It is convenient to draw the liquid to a datum point just below the indiarubber connection under the cock. The measuring burette D is graduated in cubic centimetres, and is water-jacketed to prevent fluctuations in temperature during the analysis. The burette is connected at its lower end by a flexible tube to the levelling bottle. Thus by raising the level bottle and opening the three-way cock E to discharge

through the outlet F, the water from the bottle is made to flow into the burette and so displace its gaseous contents into the atmosphere. The burette cock is now turned through a right angle to connect the burette with the horizontal tube from which the branches communicate with the various pipettes, and close the discharge to the atmosphere. In order to draw the liquid in the pipettes up to the desired level, the pipette cocks should be opened in turn, and the level bottle (being held in the hand)

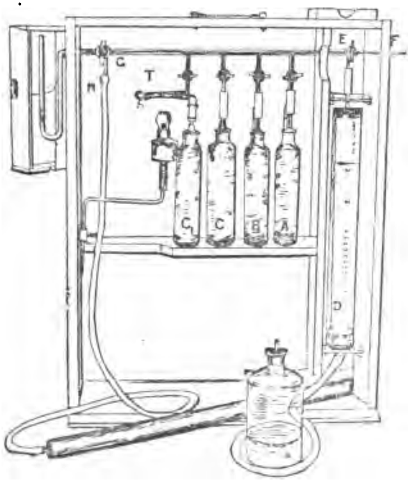


FIG. 94.

lowered until the liquid is drawn up to the datum point. This being done for each pipette, fill the burette with water up to its stop cock by raising the level bottle, and the apparatus will be ready for use. The cock G controls the inlet to the apparatus, and for convenience has a branch to the atmosphere at H. The sampling pipe leading to the producer is attached to the horizontal capillary tube by means of a $\frac{1}{2}$in. tube packed loosely with cotton wool. This acts as a filter and traps any dust that might quickly block

the capillary tubes. In the illustration a pump is shown attached to the branch H, and the controlling cock G is in the position for pumping the gases through the sampling pipe. Having made certain that all the air is drawn from the sampling pipe, and that the gas to be sampled has reached the apparatus, the pumping is discontinued, and the cock G turned through a right angle so as to afford a communication with the burette and at the same time to cut off that to the atmosphere. When the water in the burette has fallen below the zero point the cock G is again placed in the position for pumping, which motion traps the sample now drawn into the burette. In order to obtain just 100 c.c.'s of gas, the level bottle should be raised and the gases compressed up to the zero point in the burette. This done, the rubber connection should be between the bottle, and the burette should be squeezed between the fingers to retain the water in the burette, while the cock E is momentarily opened to the atmosphere to allow the excess of pressure to escape. We now have a sample of 100 cubic centimetres of gas at atmospheric pressure, and all further measurements of volume must be made at this pressure by holding the bottle so that the water in it is at the same level as that in the burette.

The level bottle is now raised and the cock opened to the pipette A, and the whole volume of gas is driven over into it. In from five to ten minutes the carbonic acid gas will be absorbed from the sample by contact with the potassium hydrate solution, and the contraction of volume is then measured by drawing the gas back to the burette until the liquid in the pipette reaches the datum point. Time should be given for the drops of water in the burette to run down, then, the level bottle being raised as above described to obtain the reading at atmospheric pressure, the height of water in the burette may be read off. In the case of Dowson gas the reading will be about 6, thus indicating that of the 100 cubic centimetres drawn into the burette 6, or 6 per cent. is the proportion of CO_2. In a precisely similar manner,

the oxygen is treated by the phosphorus. Of oxygen there should be none, and if any be found in the sample of Dowson gas there are grounds for the suspicion that the sampling pipe has leaked and admitted air. Carbonic oxide, hydrogen, and nitrogen form the remaining constitutents of Dowson gas. There will be less than 0 6 per cent. of marsh gas and olefiant gas together, but in most cases, as we have said, these may be neglected. The carbonic oxide may be determined by absorption in pipette C, but where only hydrogen and carbonic oxide be present it will be more convenient to determine their proportions by combustion at one operation instead of by absorbing the CO, which takes considerable time and is not always satisfactory. In order to carry this out a small volume of the residue is required, but it is well to leave the remainder of the sample in the oxygen pipette so that more than one combustion determination may be made. With the Orsat gear it is always advisable to use the lower part of the burette when possible, as the readings in the upper part cannot be as accurately made owing to the enlargement of the burette at its upper end. This being so, it is better to draw into the burette say 90 c.c.'s of air, after putting the whole residue into the phosphorus pipette. Having measured the 90 of air at atmospheric pressure, take in the 10 c.c.'s of residue. The mixture in the pipette having now got a supply of oxygen is combustible. The products of the combustion will be a certain volume of CO_2, resulting from the burning of the CO, and a certain volume of steam will be formed due to the hydrogen. This latter will condense and result in a diminution of volume which will afterwards be measured. The CO_2 produced will be determined by its absorption and so the proportion of each gas, namely, hydrogen and carbonic oxide, be determined.

The combustion of the mixture is effected by means of palladium asbestos placed in the combustion tube T. This tube is heated by a spirit lamp flame, and when in this condition the gas from the burette is passed through the tube

into the pipette containing merely water and acting as a reservoir. In passing through the heated palladium the combustibles in the gas are brought into close contact with the oxygen and combustion is effected. The gases are then drawn back into the burette and the diminution of volume is read off. The CO_2 formed by the combustion is then measured by absorption. The following figures will render the calculations necessary easily followed :—

PRACTICAL ANALYSIS OF SAMPLE OF DOWSON GAS IN THE ORSAT APPARATUS.

100 cubic centimetres at atmospheric pressure drawn into burette.

Reading before absorbing CO_2 = 100. After absorption, 94 ∴ CO_2 = 6 per cent.

Reading before absorbing O = 94. After absorption, 94 ∴ O = nil.

94 c.c. returned to oxygen pipette, and 90 c.c. of air drawn into burette plus 10 c.c. of residue from oxygen pipette. Reading before combustion = 100. After combustion, 95·9 ∴ contraction = 4·1. Reading before absorbing CO_2 (formed by combustion of CO) = 95·9. After absorption, 93·4 ∴ 2·5 c.c. of CO_2. Now 100 c.c. of CO will give 100 c.c. of CO_2 by combustion with 50 c.c. of oxygen ∴ it is evident that there must have been 2·5 c.c. of CO in the 10 c.c. of residue burnt. The total residue was 94 c.c. ∴ $2·5 \times \frac{94}{10}$ = total vol. of CO in 100 vols. of Dowson gas = 23·5 per cent.

In burning the CO to CO_2 a volume of oxygen equal to half the volume of CO disappears, ∴ 1·25 c.c. of contraction is due to the combustion of the CO. The total contraction was 4·1 ∴ contraction due to hydrogen and oxygen was 4·1 − 1·25 = 2·85 c.c. Of this contraction only $\frac{2}{3}$ represents volume of hydrogen present ∴ $2·85 \times \frac{2}{3} = 1·9$ vols. of hydrogen in 10 c.c. of residue. Therefore in total residue we have $1·9 \times \frac{94}{10} = 17·86$.

Collecting the results, and estimating the nitrogen by difference, we have—

CO_2	=	6·0 per cent.
CO	=	23·5 per cent.
H	=	17·8 per cent.
N	=	52·7 per cent.

CHAPTER XIV.

Calculations Required in Working Out Results of Engine Trials.

The most important deductions to be made from the analysis of the gases used are—(1) The quantity of air required for complete combustion; (2) to obtain the calorific value of the gas; (3) to calculate the specific heat of the waste gases. Of these the two first mentioned are of the greatest importance from a commercial standpoint. The loss of heat in the waste gases is difficult to determine with accuracy, and where the temperature of the gases is only approximately known it is an absurd refinement to calculate the specific heat of the gases to three places of decimals.

For the complete explanation of the reactions which take place between the constituents of coal gas and oxygen, the reader is referred to any modern elementary text-book on chemistry. A knowledge of the facts set forth in the following table enables us to work out the theoretical quantity of air required for combustion.

The nitrogen, carbon-dioxide, and oxygen are not combustible. Strictly, the oxygen contained in the coal gas should be deducted from that quantity given in column (*h*), but the oxygen is usually in such small quantities as to be quite negligible.

Air is composed of 23 per cent of oxygen and 77 per cent of nitrogen by weight; therefore the weight of air required per pound of coal gas will be

$$2{\cdot}89 \times \frac{100}{23} = 12{\cdot}5.$$

In other words, the proportion of air to gas by weight for complete combustion is equal to 12·5 to 1. One cubic foot of air weighs 0·08082 lb. Hence the proportion of air to gas by volume equals 5·72 to 1. It must be remembered that all these figures are worked out for a temperature of 32 deg.

Fah. and a pressure of 14·7 lb. per square inch absolute. Any increase of pressure causes an increase in density—that is, an increase in weight per unit volume. Conversely, any increase in temperature causes a decrease in density when the pressure remains constant—that is, decrease in the weight per unit volume.

Constituents.	(a) Volume per cent.	(b) Weight of one cubic foot.	(c) Calorific value per pound.	(d) Proportion of weight of oxygen required.	(e) Weight in one cubic foot of coal gas.	(f) Proportion by weight.	(g) Calorific value per cubic foot of coal gas.	(h) Weight of oxygen required.
		Lbs.	B. T. U.		Lbs.		B. T. U.	Lbs.
Marsh gas........	41·78	0·0447	21,690	4	0·01867	0·506	404·9	2·024
Olefines	2·77	0·1174	20,260	$\frac{24}{7}$	0·00325	0·088	65·8	0·302
Hydrogen ,.......	39·11	0 00559	52,500	8	0·00218	0·059	114·4	0·472
Carbon monoxide.	8·22	0·0783	4,300	$\frac{4}{7}$	0·00643	0·174	27·7	0·099
Nitrogen.........	8·00	0·0783	..	..	0·00626	0·170	..	..
Carbon - dioxide and oxygen...	0·12	0·1060	..	..	0·00012	0·003	..	..
	..	..	..	..	0·03691	1·000	612·8	2·897

Column (*a*) gives the volumes per cent of the constituents as furnished by analysis.

Column (*b*) gives the known weights of each constituent per cubic foot at 32 deg. Fah. and at 14·7 lb. per square inch in absolute pressure.

Column (*c*) gives the known heating value of 1 lb. of each constituent at 32 deg. Fah. and 14·7 lb. square inch absolute.

Column (*d*) gives the proportion of weight of oxygen required to combine with a given weight of gas. Thus one cubic foot of marsh gas weighs 0·0447 lb., and will require for its combustion four times that weight of oxygen; $0·0447 \times 4 = 0·1788$ lb. oxygen.

From these data the remaining columns may be filled in.

Column (*e*).—Multiply figures in column (*a*) by those in column (*b*).

Column (*f*).—This gives the proportion of weight of constituents in 1 lb. of coal gas. These figures are found by dividing the weight of each constituent in column (*e*) by the total weight of one cubic foot of coal gas.

Column (*g*).—These figures are the products of $(c) \times (e)$.

Column (*h*).—These figures are the product of $(d) \times (f)$.

The calorific value of the gas, as given in column (*g*), should agree with the experimental determination by Junker's calorimeter when both are reduced to the same standard. The calorific value for hydrogen is given minus the latent heat taken up in the formation of steam; therefore no correction need be made, as described in the text relating to Junker's calorimeter.

It is sometimes of interest to calculate the maximum temperature possible during the combustion of the mixture in the cylinder. To do this we require to know the thermal value of the charge of gas, the specific heat at constant volume, and the weight of the whole charge. The volume of gas entering the cylinder per cycle is easily obtained from the gas-meter readings. We will now proceed to find the specific heats of various mixtures of coal gas and air for the sample tabulated in the above table. First, to find the specific heat of the coal gas. The following table gives the data required:—

	(*a*) Proportion by weight.	(*b*) Specific heat of constituent at constant volume.	(*c*)
Marsh gas	0·506	0 470	0·2378
Olefines	0·088	0·332	0·0292
Hydrogen	0·059	2·406	0·1419
Carbon-monoxide	0·174	0·173	0·0301
Nitrogen	0·170	0·173	0·0294
Carbon-dioxide	0·003	0·171	0 0048
Oxygen	....	0·155	
			0·4732

Column (*a*) gives the proportion by weight of the constituents of the coal gas. These figures are copied from the previous table.

Column (*b*) gives the specific heats of the constituents at a constant volume.

Column (*c*).—These figures are the result of the product of the figures in column (*a*) by column (*b*), and the total is the specific heat of the gas. This equals 0·473.

Taking the specific heat of air as 0·168 at constant volume, we will next work out the specific heats of the following mixtures of air and gas :—

Volume of air to gas.	Specific heat of mixture at constant volume.	Proportion of air to gas by weight.
6—1	0·189	13·13 to 1
7—1	0·186	15·32 to 1
8—1	0·184	17·51 to 1
9—1	0·183	19·70 to 1
10—1	0·181	21·89 to 1
11—1	0·180	24·07 to 1
12—1	0·179	26·26 to 1
13—1	0·178	28·45 to 1
14—1	0·178	30·64 to 1
15—1	0·177	32·83 to 1

First find the proportion of air to gas by weight. The ratio of the weight of one cubic foot of air to one of gas

$$= \frac{0{\cdot}0808}{0{\cdot}0369} = 2{\cdot}189.$$

Therefore proportion of 6 to 1 mixture by weight = 6 × 2·189 to 1 = 13·134 to 1. The other figures are similarly calculated.

Proceeding now to obtain the specific heat of a 6 to 1 mixture, we have (proportion by weight of air) × (specific heat of air) + (proportion by weight of gas) × (specific heat of gas) ÷ (total weight of air + gas). This, in figures,

$$= \frac{(13{\cdot}13 \times 0{\cdot}168) + (1 \times 0{\cdot}473)}{13{\cdot}13 + 1} = \frac{2{\cdot}205 + 0{\cdot}473}{14{\cdot}13} = \frac{2{\cdot}678}{14{\cdot}13} = 0{\cdot}189.$$

Similarly the remaining figures in the above table are arrived at.

The specific heat of any mixture may be determined approximately by a formula known as Grashof's formula.

The formula is based upon a mean specific heat for coal gas, assuming that its constituents never vary in their proportion one to the other.

Let c_v = specific heat of mixture at a constant volume.
R = ratio of vols. of air to gas in the mixture;

then
$$c_v = \frac{\mathrm{R} \times 0{\cdot}168 + 0{\cdot}226}{\mathrm{R} + 0{\cdot}415}.$$

Putting in values for the above case, already worked out from the exact data, we obtain,

$$\begin{aligned} c_v &= \frac{\frac{6}{1} \times 0{\cdot}168 + 0{\cdot}226}{\frac{6}{1} + 0{\cdot}415} \\ &= \frac{1{\cdot}234}{6{\cdot}415} \\ &= 0{\cdot}192, \end{aligned}$$

the error in this instance amounting to about $1\frac{1}{2}$ per cent.

The following will serve as an example of the use of these figures, until we come to consideration of the indicator diagram.

Suppose 6 cubic feet of air to be mixed with 1 cubic foot of the coal gas given in the previous analysis. What would be the maximum temperature possible when the explosion takes place at 32 deg. Fah. and 14·7 lb. pressure?

Data required.—(1) Specific heat of mixture; (2) weight of mixture; (3) calorific value of the gas.

Then temperature

$$\begin{aligned} &= 32 + \frac{\text{total heat present}}{\text{weight of mixture} \times \text{specific heat}} \\ &= 32 + \frac{612}{0{\cdot}521 \times 0{\cdot}189} \\ &= 32 + 6320 \text{ (nearly)} \\ &= 6352 \text{ deg. Fah.} \end{aligned}$$

In a gas engine the theoretical temperature is never recorded by the indicator card, as there is a very rapid transmission of heat through the walls of the cylinder.

CHAPTER XV.

Calculations on the Indicator Diagram.

The Indicator Diagram.—We have already considered the means and method of actually taking an indicator diagram. It now remains to be shown what results may be obtained from it by simple calculations. The indicator diagram furnishes information upon the following points, which we will consider in order: (1) The indicated horse power; (2) the valve setting; (3) the initial, maximum, and exhaust temperatures; (4) the nature of the expansion and compression curves.

Although the indicated horse power of a gas engine is never a very certain quantity, it is always calculated in gas-engine trials. The inertia of the indicator levers, the effects of high temperature upon the indicator spring, and the difficulty in obtaining a true average from a number of diagrams when the engine is running below its full load, all contribute towards inaccuracy. For these reasons the ratio of brake horse power to indicated horse power cannot always be relied upon as correct. High mechanical efficiencies of gas engines must, therefore, be regarded with suspicion unless we have complete access to all the details of the test.

The indicated horse power is found as follows:—

Let I = the indicated horse power.
P = mean pressure per square inch on the piston.
L = length of stroke in feet.
E = number of explosions per minute.
A = area of cylinder in square inches.

Then

$$I = \frac{L\,E\,A\,P}{33000}.$$

In working out a number of diagrams from one engine, it will be found convenient to work out in decimal form the

fraction $\frac{L\,A}{33000}$, and record this once for all as the engine constant.

The factor we require to determine from the diagram is the mean pressure per square inch. Vertical measurements give pressures, and horizontal measurements distances; hence the length of an ordinate measured with the scale of pressures gives the pressure per square inch at a particular point of the stroke. If the pressure remained constant throughout the whole stroke, the diagram would become a rectangle, and the product of pressure and distance would give the work done. From this we see that the length × breadth, or area of the diagram, represents to some scale the work done per stroke. We may therefore find the mean pressure upon the piston, either by getting the average height of a set of equidistant ordinates, or by dividing the area of the diagram by the length.

To facilitate the work of finding the mean height, instruments commonly known as averagers are used. Before illustrating these we will briefly describe the most expeditious way of calculating the mean height without special instruments.

Divide the diagram into ten equal vertical strips. Subdivide each of these by a dotted line passing through their centres. It is required to find the average height of these dotted lines. Take a strip of paper, and mark off upon its edge the length of the first ordinate, then add to this the length of the second by applying the paper to the second ordinate. In this way we obtain the sum of the lengths of the ordinates. Measure this with a rule divided into inches and tenths. Suppose this gives us 5·08 in., we know that the average height will be 0·508 in. If the diagram be taken with a 100 spring, our mean pressure becomes 50·8 lb. per square inch. When a number of diagrams are to be dealt with, it will be found convenient to construct a set of ten converging lines upon tracing paper. By applying this to

the indicator diagrams, the ten ordinates may readily be pricked off.

Where a great deal of testing is done, the work will be much facilitated by the use of an averager for obtaining mean pressures. Of these we shall describe very briefly the two most commonly used, viz., the Goodman and Coffin averagers.

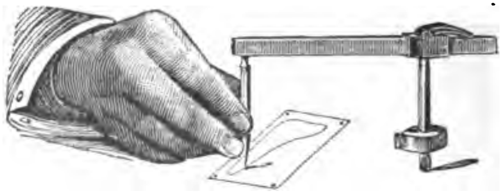


FIG. 95.

Goodman's averager is shown in fig. 95, and is supplied by Messrs. Jackson Brothers, Leeds. The tracing point is fixed to the horizontal bar. The other leg of the instrument is carefully ground to a knife edge in such a way that the edge, if produced, would pass through the tracing point. The distance between the knife edge and the tracing point is adjustable by sliding the former along the horizontal bar, by means of which the instrument is set to the exact length of the diagram to be measured. Suppose the diagram, fig. 96, is to be measured. Choose a point A somewhere about the centre of the figure, and draw any line A B to meet the boundary. The tracing point is placed upon A, and the the hatchet end put so that the mean position of the instrument is roughly square with A B, as shown at X. The position of X is marked by pressing the knife edge upon the paper. Now take the tracing point lightly in the fingers, so that the movement of the hatchet end is not controlled by the pressure of the hand. Travel from A to B, then follow the direction of the arrows round the boundary of the figure, say in a clockwise direction, returning to the point B, thence back to A. The knife edge will then occupy a new

position at Y, which is marked as before. The indicator diagram should now be turned about the point A through 180 deg. The figure is again traced by the point, but this time in a contra-clockwise direction, following the dotted outline, fig. 96. When the point returns again to A the hatchet will have travelled back towards X, say to X_1. The mean side movement of the hatchet end gives at once the average height of the diagram. Thus, suppose the distance from Y to the dot between X and X_1 is measured off as 0·56 in.; then this is the average height of the diagram, and multiplying 0·56 by the scale of pressures gives the mean pressure upon the piston.

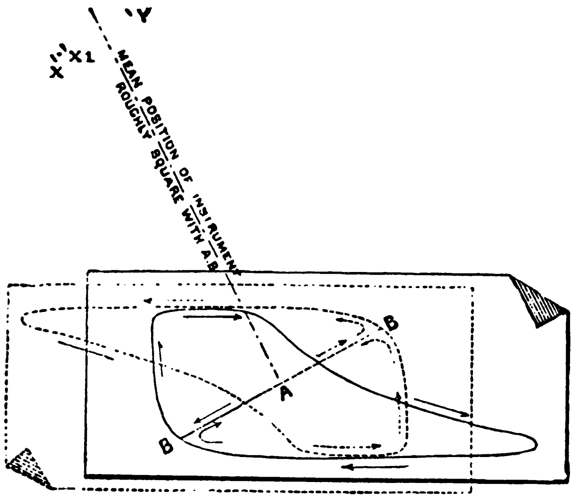


Fig. 96.

The ordinary hatchet planimeter was invented by Captain Prytz, of Copenhagen, but its use involved calculations far too tedious for practical men. Professor Goodman has embodied these calculations on a scale marked upon his hatchet planimeter, which enables an area to be measured by reading off the side travel of the hatchet end as so many square inches

marked upon the scale. The averager described is so arranged that the area of the figure traversed by the boundary is equal to the side travel of the hatchet multiplied by the length of the diagram. The complete theory of these instruments is beyond the province of this work ; we may, however, refer those interested to the pages of *Engineering*, vol. lvii., page 687, also vol. lxii., page 225.

Before leaving the subject, it is is well to point out that accuracy can only be obtained by careful manipulation. The instrument should be carefully set to the total length of the diagram to be measured. The hatchet end should travel on a good surface, such as drawing or blotting paper, and, finally, the diagram should always be reversed so that the *mean* travel of the instrument may be read off. With these precautions Goodman's averager will be found valuable in quickly determining the average pressures. It further possesses the important advantages that it is not liable to derangement by rough usage, and it is procurable at a price far below that of any other instrument used for the same purpose.

The Coffin averager supplied by the Globe Engineering Company, Manchester, is illustrated in fig. 97. This instrument consists of a bar, carrying a recording wheel, having a tracing point at one end, and a pin at the other end, moving in a vertical slot. The only movable parts are the bar, with its wheel and the vertical sliding piece, marked K. The diagram to be measured is placed upon the instrument as shown in the illustration. The atmospheric line should be parallel with the edge of the square marked B. The vertical edges, marked C and K, should nearly touch the extremities of the diagram. The tracing point D, when moved down the inner edge of K, should then coincide with the extremity of the diagram. The weight W is intended to keep the instrument from lifting out of the slot when in use. The recording wheel runs upon a piece of paper fastened to the board upon which the other parts are mounted.

Having placed the diagram in position, make an indentation at its extreme end, as at E, with the tracing point D, at the same time setting the wheel to zero. Then trace out

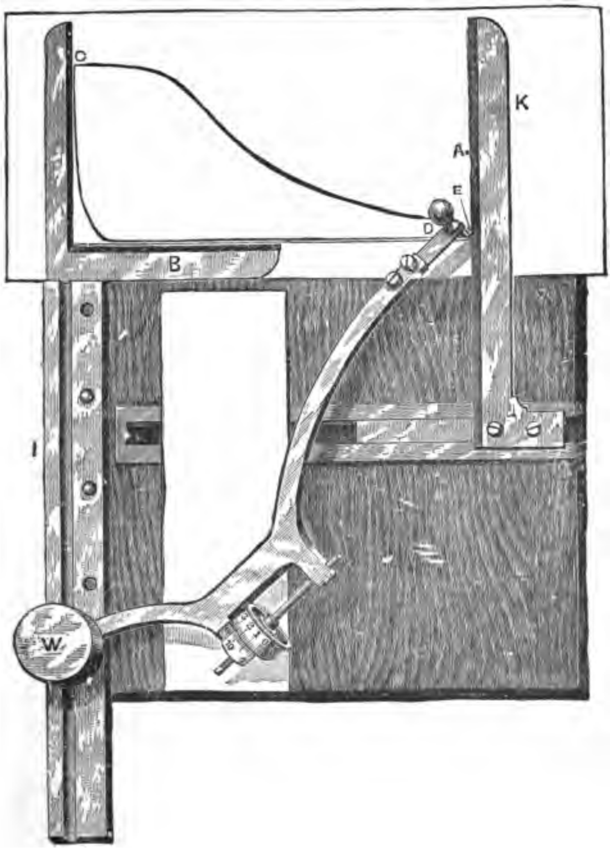


FIG. 97.

the diagram by moving in a *clockwise direction*, until returning to the point E. The *area* of the diagram may now be

read off in square inches upon the wheel and its vernier scale. We, however, require the mean height of the diagram; hence, if we move the pointer D upwards along the inner edge of K until the wheel returns to zero, this vertical distance E A will give the mean height. This distance should be measured with the scale of pressures corresponding to the spring used in taking the diagram. The distance A E, as found by this instrument, would be the same as the mean-side travel of the Goodman averager, and might be measured in inches, and then multiplied by the scale of the spring.

The well-known Amsler's planimeter is often used for determining the area of indicator diagrams. It is, however, much more convenient to use one of the instruments above described, as they are specially designed for the purpose, whereas Amsler's planimeter is designed for measuring much larger areas than usually obtained on indicator diagrams.

We have already incidentally touched upon the defects of diagrams occasioned by bad valve setting. A weak-spring diagram is necessary to show the action of the inlet valves. These valves, being too small or not having sufficient lift, will cause the suction line of the diagram to fall below the atmospheric line. Late admission of the inlet valves would, of course, be shown by a marked depression of the suction line at the early part of the stroke. When the valves opened the suction line would rise towards the atmospheric.

With respect to the exhaust, a frequent fault is its late opening. This is invariably caused by the wear of the spindle. An adjusting screw is provided on the lifting lever, so that the lead of the exhaust valve may be increased. The effect on the diagram is to give a sharp point to the toe, with a falling exhaust line.

The next point to be considered is the temperature of the gases in the cylinder during the working stroke. We have already stated that without very special apparatus the

temperature of the exhaust gases cannot be measured experimentally. It is, however, customary to furnish in reports of engine trials an account of the heat distribution. For this reason we require to know the temperature of the exhaust gases, so that we may determine approximately the heat thrown away as the exhaust gases pass into the atmosphere.

It is usually assumed that the following law holds under the conditions in a gas-engine cylinder. When P = absolute pressure, V = volume, and T = absolute temperature, we have

$$\frac{P\,V}{T} = \text{constant.}$$

If, therefore, we determine one set of values for P, V, and T, we can calculate corresponding values for any point on the diagram. The values of P and V are easily measured from the diagram, but at no time is the value of T actually known. We do know, however, the temperature of the jacket water, and it is certain that the inner walls of the cylinder will be hotter than the jacket water. Then, again, exhaust gases left in the clearance volume of the cylinder will help to raise the temperature of the cold incoming mixture. As a compromise, it may be assumed that the temperature of the incoming mixture, before compression commences, is 5 deg. Fah. above that of the jacket water.

Suppose the following data obtained: Engine clearance, 30 per cent; length of indicator diagram, 3·5 in.; pressure during the instroke obtained from weak-spring diagram, 13·5 lb. absolute per square inch; temperature of jacket water, 150 deg. Fah. Let the exhaust take place at 0·9 of the stroke, and let the pressure measured off from the atmospheric line at the moment of exhaust be 25 lb.

Horizontal measurements on the indicator diagram represent distances travelled by the piston; but the piston diameter being constant, these same measurements may also be taken to represent volumes swept out by the piston; 3·5 in. therefore represents the working volume of the

cylinder. The clearance volume is known to be 30 per cent of this. ; thus the whole volume is represented by a length of

$$3{\cdot}5 + 3{\cdot}5 \times \frac{30}{100} = 4{\cdot}55 \text{ in.}$$

$$\frac{P\,V}{*T} \text{ then} = \frac{13{\cdot}5 \times 4{\cdot}55}{150 + 5 + 460} = 0{\cdot}0982.$$

Value of V when exhaust valve opens may be measured from the diagram. In our data it becomes

$$(3{\cdot}5 \times 0{\cdot}9) + \left(3{\cdot}5 \times \frac{30}{100}\right),$$

which equals 4·15. The value of P in pounds per square inch absolute = 25 + 14·7 = 39·7 ;

therefore
$$T = \frac{39{\cdot}7 \times 4{\cdot}15}{0{\cdot}0982}$$
$$= 1677 \text{ absolute}$$
$$= 1217 \text{ deg. Fah.}$$

The heat thrown away in the exhaust gases per minute will therefore be (temperature Fah.) × (weight of products of combustion per minute) × (specific heat at a constant volume of products of combustion).

The specific heat at a constant volume, and at constant pressure, may be found by Grashoff's formulæ. When R = ratio of air to gas by volume,

specific heat of products at constant pressure

$$= \frac{0{\cdot}2375 \times R + 0{\cdot}343}{R + 0{\cdot}48};$$

specific heat of products at constant volume

$$= \frac{0{\cdot}1684 \times R + 0{\cdot}286}{R \times 0{\cdot}48}.$$

The temperature at any point on the indicator diagram may be found as above. In calculating the maximum temperature, as shown on the indicator diagram, a correction

* Absolute temperature = temperature in deg. Fah. plus 460.

for an increase of volume should be made if the explosion line be other than perpendicular.

The maximum temperature of the gases as shown on the indicator diagram is little more than half the theoretical temperature possible. This is partly owing to the rapid transmission of heat through the walls of the cylinder to the jacket, and partly to the fact that the whole of the heat is not evolved when the highest pressure is reached. From certain characteristics of the expansion curves it is probable that the burning of the gases continues after the maximum pressure is arrived at.

We have now discussed the deductions which are usually required to be made from the indicator diagram when reporting a gas-engine trial. There are many other points of scientific interest. These are, however, rather beyond the scope of the present work, but are completely discussed in a book recently published upon "Heat and Heat Engines," by Mr. W. C. Popplewell.

Before leaving the subject of the indicator diagram, it may be well, for the sake of completeness, to state certain facts with respect to the expansion and compression curves, leaving those interested to pursue the question further.

The equation to the expansion and compression curves is of the form $P V^n =$ constant. When the value of n is equal to

$$\frac{\text{specific heat of products of combustion at constant pressure}}{\text{specific heat of products of combustion at constant volume}}$$

the expansion is adiabatic—that is, without loss or gain of heat through the cylinder walls. In a gas engine the water jacket is constantly abstracting heat from the walls; hence we should expect to find the expansion curve much below the adiabatic. This, however, is not the case; indeed, in practice it is found that the expansion curve is sometimes above the adiabatic, though usually slightly below. In view of the fact that the maximum temperature of the explosion is little more than half the theoretical temperature possible, and yet the expansion curve so nearly follows the

adiabatic line, notwithstanding the rapid loss of heat to the cylinder walls, it must be concluded that burning continues long after the maximum pressure has been reached.

The compression curve, as might be expected, is very nearly adiabatic.

CHAPTER XVI.

Gas-engine Trial, Otto Cycle: Coal Gas Used.

We have now explained the apparatus required for a complete test of a gas engine, and we have discussed various calculations in detail. It now remains for us to go through the figures of a complete trial, which may be taken as a typical example of modern practice.

Measurements of Engine.—

Diameter of cylinder, 9 in.; stroke, 18 in.
Clearance volume of cylinder, 490 cubic inches.
Diameter of brake wheel, 4 ft.
Diameter of brake rope, 0·75 in.
Weight of unbalanced part of brake rope, including hook, 5 lb.

Note.—Spring balance tested by dead weights, and found to be reading 2 lb. in excess of real weight.

Mean Results of Observations taken during a Two Hours' Test.—

Brake readings { Spring balance (uncorrected), 17·5 lb.
Load on brake (excluding hook), 255 lb. }

Mean revolutions of engine by counter per minute, 160.

Explosions per minute, 80.

Gas-meter readings { Total gas in cubic feet, 815.
Temperature of meter, 65 deg. Fah.
Pressure in mains, 0·9 in. water. }

Jacket water $\begin{cases} \text{Quantity (total), 1,860 lb.} \\ \text{Inlet temperature, 58 deg. Fah.} \\ \text{Outlet temperature, 155 deg. Fah.} \end{cases}$

Mean pressure on piston derived from indicator diagrams, 80·5 lb. per square inch.

Pressure when exhaust valve opens, 41 lb. per square inch.

Pressure at end of suction stroke, 13·8 lb. per sq. inch absolute.

Exhaust commences at 88 per cent of working stroke.

Barometric pressure, 15 lb. per square inch.

Calorific value of gas per cubic foot obtained by Junker's calorimeter at a pressure of 0·9 in. water = 615 B.T.U.

Temperature of calorimeter meter, 65 deg. Fah.

Calculation of Results from above Data.

$$\text{I.H.P.} = \frac{9^2 \times 0{\cdot}7854 \times 1{\cdot}5 \times 80{\cdot}5 \times 80}{33000} = 18{\cdot}6.$$

B.H.P. Corrected spring balance reading = 17·5 − 2 = 15·5 lb.

Total load on brake = 255 + 5 = 260 lb.

Net load on brake = 244·5 lb.

Effective diameter of brake wheel $= 4 + \frac{0{\cdot}75}{12}$ ft. = 4·062 ft.

$$\text{B.H.P.} = \frac{4{\cdot}062 \times 3{\cdot}14 \times 160 \times 244{\cdot}5}{33000} = 15{\cdot}1.$$

$$\text{Mechanical efficiency} = \frac{\text{B.H.P.}}{\text{I.H.P.}} = \frac{15{\cdot}1}{18{\cdot}6} = 0{\cdot}812.$$

$$= 81{\cdot}2 \text{ per cent.}$$

Gas Used per I.H.P. Hour.—Total consumption of gas uncorrected for pressure and temperature = 815 cubic feet. Consumption reduced to 32 deg. Fah. and 14·7 lb. per square inch,

$$= 815 \times \frac{32 + 460}{65 + 460} \times \frac{15 + (0{\cdot}434 \times 0{\cdot}075)}{14{\cdot}7}$$

= 781 cubic feet (corrected).

Gas used per I.H.P. hour (reduced to 32 deg. and 14·7 lb.)

$$= \frac{781}{2 \times 18.6} = 21 \text{ cubic feet.}$$

Gas used per B.H.P. hour (reduced to 32 deg. and 14·7 lb.)

$$= \frac{781}{2 \times 15.1} = 25.9.$$

The ratio of air to gas should be determined from an analysis of the products of combustion. When not done in this way, account must be taken of the temperature and pressure of the gas at the end of the suction stroke. The temperature is difficult to determine with precision, but is usually assumed to be equal to that of the outlet jacket water. Under this assumption the following figures give the ratio of air to gas by volume.

Volume of gas per cycle (corrected for temperature and pressure at end of suction stroke)

$$= \frac{781}{80 \times 2 \times 60} \times \frac{14.7}{13.8} \times \frac{155 + 460}{32 + 460} = 0.108 \text{ cubic feet.}$$

Working volume of cylinder + clearance volume $= 0.947$ c. ft.
Volume of air per cycle $= 0.947 - 0.108 = 0.839$ cubic feet.

Ratio of air to gas $= 7.7$ to 1.

NOTE.—When the engine is worked without a scavenging device the working volume only is considered.

Specific heat of products of combustion at constant volume (by Grashoff's formula)

$$= \frac{0.168 \times 7.2 + 0.286}{7.2 + 0.48} = 0.196.$$

Temperature of Exhaust Gases, by Calculation from Indicator Diagram.—Temperature of gases before compression commences, assumed as temperature of jacket water + 5 deg.,

$$= 155 + 5 = 160.$$

Total volume of cylinder = clearance volume + working volume

$$= 490 + 1145 = 1635 \text{ cubic inches,}$$
$$= 0.948 \text{ cubic feet.}$$

$$\frac{P\,V}{T} = \frac{P^1\,V^1}{T^1}$$ when P = pressure (absolute).
V = volume (total).
T = temperature (absolute).

Filling in values—

P = 13·8 lb. per square inch absolute.

V = 0 948 cubic feet.

T = 160 + 460 = 520.

P^1 = pressure when exhaust valve opens = 14·7 + 41.

V^1 = (1145 × 0 88) + 490 = 0 867 cubic feet.

Then

$$T^1 = \frac{T\,P^1\,V^1}{P\,V} = \frac{55{\cdot}7 \times 0{\cdot}867 \times 520}{13\,8 \times 0\,948} = 1920.$$

Temperature = 1920 − 460 = 1460 deg. Fah.

Heat Lost in Exhaust Gases.—Weight of 1 cubic foot of coal gas = 0·0369. Air weighs 2 63 more than coal gas. Therefore weight of charge per cycle

$$= 0{\cdot}948 \left\{ \frac{0{\cdot}0369 \times 2{\cdot}63 \times 7\,2}{8\,2} + \frac{0{\cdot}0369}{8\,2} \right\} = 0{\cdot}081 \text{ lb.}$$

Heat lost in exhaust gases per cycle

$$= 0{\cdot}081 \times 1460 \times 0{\cdot}196 = 231 \text{ B.T.U.}$$

Heat lost in exhaust gases per minute

$$= 231 \times 80 = 1848 \text{ B.T.U.}$$

Heat lost in jacket water per minute

$$= (155 - 58) \times \frac{1860}{2 \times 60} = 1503 \text{ B.T.U.}$$

Heat equivalent of work done in the cylinder

$$= \frac{18\,6 \times 33000}{772} = 793 \text{ B.T.U.}$$

Heat equivalent of work done on brake

$$= \frac{15{\cdot}6 \times 33000}{772} = 644 \text{ B.T.U.}$$

Heat Received by Engine per Minute.—Calorific value of gas, as given by calorimeter, and corrected for hydrogen only = 615 B.T.U. Correcting for pressure and temperature, we have—

Calorific value at 32 deg. Fah. and 14·7

$$= 615 \times \frac{525}{492} \times \frac{14 \cdot 7}{15 \cdot 03} = 640 \text{ B.T.U.}$$

Heat received per minute by engine

$$= 640 \times \frac{781}{2 \times 60} = 4165 \text{ B.T.U.}$$

Heat efficiency, as calculated on equivalent of work done in cylinder,

$$= \frac{793}{4165} = 0 \cdot 186 = 19 \text{ per cent.}$$

Heat efficiency, as calculated on work done on brake,

$$= \frac{644}{4165} = 0 \cdot 151 = 15 \cdot 4 \text{ per cent.}$$

HEAT ACCOUNT.

Heat received by engine per minute..............	4,165	Thermal equivalent of work done	793
		Heat lost in jackets...	1,503
		Heat lost in exhaust gases	1,848
	4,165		4,144

It is not unusual to find that the heat account gives a slight *excess* of heat accounted for. The temperature of the exhaust has been calculated at the point of opening of the valve—that is, at 88 per cent of the working stroke. Some of this heat is transmitted to the jacket water, and is therefore measured twice. On the other hand, large losses due to radiation are not measured. As before stated, the temperature of the exhaust gases is a rather uncertain quantity. It therefore not infrequently works out that a percentage

of heat is unaccounted for on one or other side of the heat account.

During the progress of a trial it is extremely useful to plot all quantities as they are observed upon a large sheet of squared paper. Errors of observation are thus easily discovered in time to be rectified. A break in the plotted curves will indicate either a mistake in the readings or fluctuations in the conditions of the trial. It is important that either should be checked. The person responsible for the trial will learn more from a casual glance at these curves than by attempting to check each observer individually.

CHAPTER XVII.

Gas-engine Design.

In writing upon the subject of gas-engine design, it will be assumed that the general arrangements of a gas engine are already familiar to the reader. Hitherto the calculations required to determine the sizes of the various essential parts have been entirely excluded from the literature upon the subject. Considering the difference existing between the gas engine and the steam engine, it is important that calculations determining the proportions of the former should be dealt with quite independently. We propose, therefore, to work out, from data already established, the leading dimensions of a gas engine to develop 20 horse power on the brake.

The mechanical efficiency of a gas engine may reasonably be supposed to reach 80 per cent. This is, of course, sometimes exceeded, but it is wise to underestimate, as no engine will develop its highest efficiency unless perfectly adjusted, and in the exigencies of practice this should not be relied

upon. If, therefore, we assume 80 per cent mechanical efficiency, we shall require a cylinder capable of developing $\frac{20}{0 \cdot 8} = 25$ indicated horse power.

In steam-engine design, our next subject for consideration would be the maximum boiler pressure at our disposal. From this, with due regard to the number of expansions permissible, we should ultimately draw a prospective indicator diagram, and so obtain the mean effective pressure per square inch of piston area. Now, the pressure obtained in a gas-engine cylinder depends upon three factors: (1) the mixture of air and gas; (2) the quality of the gas; (3) the density of the mixture before ignition takes place. A fourth factor—namely, the mean working temperature of the cylinder—would, if variable, greatly affect the pressure. The temperature is, however, necessarily reduced by the presence of the jacket water to an almost constant limit. Early designers of gas engines did not attempt high compression; thus, up to the year 1890, the compression seldom exceeded 40 lb. per square inch. This, however, has been gradually raised, until now the compression on small engines amounts to over 90 lb. per square inch. It is inadvisable to exceed, or even approach, this figure in designing large power gas engines, say over 50 I.H.P., on account of the liability to ignition during the compression stroke.

This risk is, however, much reduced when a scavenging arrangement clears the hot gases from the combustion chamber, and replaces them with cooler air. In round numbers, we may take it that the maximum pressure obtained will be 3·5 times the pressure before ignition. Thus, compressing up to 60 lb. per square inch, we may expect, with the usual proportions of air to gas, a maximum pressure of 210 lb. Although this is the maximum pressure obtainable when running at full power, it must not be forgotten that when the governor cuts out the gas supply

for one or perhaps more cycles, a particularly dense mixture may be drawn into the cylinder. The ignition of a strong mixture may thus produce a maximum pressure far in excess of that calculated. For this reason a large margin is necessary.

There is a marked similarity between all indicator diagrams from gas engines, and from a comparison of a large number of diagrams it will be found that the mean effective pressure produced is roughly equal to $2C - 0{\cdot}01\,C^2$ when C = compression in pounds per square inch above atmospheric pressure. These figures may be varied one way or other by the valve setting and cylinder proportions.*

The expansion curve of an indicator diagram may be raised by decreasing the time of expansion and by reducing the cylinder surface to a minimum. The first mentioned condition reduces the *time* during which heat may be transferred from the burning gases to the jacket water, and the second condition reduces the surface by means of which the transmission of heat is facilitated. Let us first consider the effects of cylinder proportions, and engine speed, upon the rate of expansion, and in so doing we will assume that the diameter of the cylinder and power developed remain constant, whilst the revolutions per minute and the length of stroke are variable.

The number of feet travelled by the piston per minute is limited. For practical reasons it is unadvisable to exceed 700 ft. per minute. When L = length of stroke in feet, and R = revolutions per minute, then we have the limiting value of piston speed

$$2\,L\,R = 700 \text{ ft.}$$

* It will be observed that this formula gives a maximum mean pressure of 100 lb. per square inch when the compression reaches 100 lb. It has been shown by Messrs. Mallard and Le Chatelier that the rate of cooling follows the law expressed by $\alpha\,\theta + \beta\,\theta^2$ where α and β are constants, and θ = temperature. It is interesting to note that the above formula, deduced from actual indicator diagrams, shows that the mean effective pressure follows a similar law. The formula holds *only up to 100 lb.* compression, and must only be regarded as giving approximate values, for neither the mixture nor quality of gas form factors of the expression.

It is evident that a variety of dimensions might be chosen for L and R. As R is increased, L is diminished. As R is increased the number of impulses per minute may be increased, and, consequently, the time of each expansion is diminished. The volume of gas used per cycle is diminished, but the maximum pressure obtained need not be, if the same compression be given to the mixture. Thus we see that a smaller volume of gas gives the same maximum pressure per square inch as a larger volume, and, further, it is expanded in very much less time. If R be increased continuously, and L correspondingly diminished, then at some value of R the outstroke of the piston will take place in less time than the charge can be effectually ignited. Let us investigate the limit of speed of any engine in which the *connecting rod is five times the crank* length.

Let T = time required for explosion pressure to arrive at its maximum;
C = length of crank radius in feet;
R = revolutions per minute of engine;

then mean speed of crank pin

$$= \frac{2\pi C R}{60} \text{ ft. per second.}$$

The mean velocity of the piston during the first one-tenth of its forward stroke will be very nearly

$$= \frac{2\pi C R}{60} \times 0{\cdot}32 \text{ ft. per second.}$$

The distance travelled

$$= \frac{\text{stroke}}{10} = \frac{2\,C}{10} = 0{\cdot}2\,C \text{ feet.}$$

Therefore time taken by piston in travelling one-tenth stroke

$$= \frac{0{\cdot}2\,C}{\frac{2\pi C R \times 0{\cdot}32}{60}}$$

$$= \frac{5{\cdot}9}{R} \text{ (nearly) seconds.}$$

Now, in order that the engine may fully expand the hot gases, the maximum pressure should not be reached at a point later than one-tenth of the stroke. Even so late as this is disadvantageous. Hence the lowest value of

$$\frac{5 \cdot 9}{R} \text{ must equal T.}$$

Mr. Dugald Clerk ascertained that the time required to reach the maximum pressure, with a mixture of coal gas and air in the proportion of 1 to 5 by volume, was 0·05 second. In his experiments the initial temperature of the mixture was low, and the initial pressure was atmospheric. The author has found that the time of explosion is greatly diminished by raising the initial pressure, but has not yet succeeded in measuring accurately by how much. Witz found that the duration of explosion of a mixture of 1 to 6·3, behind a freely-moving piston, to be 0·06 second. As the ignition took place at atmospheric pressure, these results can hardly be applied to modern gas engines. It is probable that the maximum number of revolutions approaches the limit at 500 per minute. Putting this value for R in the equation,

$$\frac{5 \cdot 9}{R} = T,$$

we find $T = 0 \cdot 0118$ second. We might here appropriately consider the effect of *lead* on the igniting valve. The total time of rise of pressure in a gas-engine cylinder may be divided into two distinct parts. Firstly, the time taken for the flame to strike back into the mixture, and, secondly, the time during which the pressure rises after this has been accomplished. The former effect may be largely compensated for by giving lead to the ignition valve, but the latter cannot be dealt with in this way without seriously increasing the liability to what is known as "back explosion." The severe strains occasioned by such back explosion should, of course, be avoided. In the absence of experimental data with regard to the values of T, we shall do well to accept the limiting speed of revolutions as

approaching 500. In practice it is found advisable to run large engines at a speed much below 500 (about 160) revolutions, because of the excessive vibration due to the rapid movements of reciprocating parts, and the consequent stresses brought to bear upon them. It is also probable that the value of T is much increased when large cylinders are used. A large number of smaller engines run at from 250 to 300 revolutions.

The result of our investigation has, so far, led to the conclusion that the speed approaches a limit at 500 revolutions per minute, and it has been further shown that this is independent of the length of stroke, on the assumption that the maximum pressure shall be reached during the first tenth of the stroke. We have now to consider the ratio of length of stroke to diameter of cylinder.

It is certain that the loss of heat increases in a greater proportion than the difference in temperature between the burning gases and the cylinder walls. Also the loss of heat is greater, the greater the density of the charge. In other words, the loss of heat is very much greater during the beginning of the stroke than at any other time. If it be true that the loss of heat varies directly as the density, and directly as the surface, and roughly as the difference in temperature, then we may express the loss of heat in a time t_x as

$$A\int_0^{t_x}(\text{surface} \times \text{difference in temperature} \times \text{density})\,dt.$$

As, however, the density varies inversely as the volume, we may write the expression thus—

$$B\int_0^{t_x}\left(\frac{\text{surface} \times \text{difference in temperature}}{\text{volume}}\right)dt,$$

A and B being constants.

From this it is evident that the ratio of surface to volume should be a minimum near the beginning of the stroke. To

investigate the subject further, by attempting to evaluate the quantities in the above expression, would be rather begging the question, inasmuch as the temperature differences must be estimated by references to the diagrams taken from existing engines. Experiments might be devised to furnish data upon this subject, but the author is not aware that such independent data at present exists.

It is the opinion of Mr. Hamilton, the patentee of the Premier gas engine, that the ratio $\frac{\text{surface}}{\text{volume}}$ should be a minimum at about one-third of the stroke. Supposing, therefore, that the combustion chamber were one-third the length of the stroke, then we should require a minimum ratio of $\frac{\text{surface}}{\text{volume}}$ when the volume is actually equal to (area of piston × two-thirds stroke). Now, it is easily shown that the ratio $\frac{\text{surface}}{\text{volume}}$ is a minimum when the diameter of a cylinder is equal to its length.* Thus we see that a minimum ratio of

---

* Let x = diameter, c = volume (constant). Then, length

$$= \frac{4\,c}{x^2\,\pi}.$$

Surface $$= \frac{x^2\,\pi}{2} + \frac{4\,c}{x^2\,\pi}\,.\,x\,\pi$$

$$= \frac{\pi}{2}\,x^2 + 4\,c\,\frac{1}{x}$$

$$d\,\mathrm{S} = \frac{\pi}{2}\,d\,(x^2) + 4\,c\,d\left(\frac{1}{x}\right) = \frac{\pi}{2}\,2\,x\,d\,x - \frac{4\,c}{x^2}\,d\,x$$

$$\frac{d\,\mathrm{S}}{d\,x} = \pi\,x - \frac{4\,c}{x^2}.$$

When surface is a minimum $\frac{d\,\mathrm{S}}{d\,x} = 0.$

$$\therefore\ \pi\,x^3 - 4\,c = 0\ \therefore\ 4\,c = x^3\,\pi.$$

But length $$= \frac{4\,c}{x^2\,\pi},$$

and, by putting $4\,c$ in terms of x, we have length

$$= \frac{x^3\,\pi}{x^2\,\pi} = x.$$

Hence length = diameter when surface is a minimum.

$\frac{\text{surface}}{\text{volume}}$ is given when the diameter is = two-thirds stroke. This, as may be supposed, is a favourite proportion. The author is convinced that the effect of large cylinder surface is very marked.

Cylinders of large dimensions have a much larger proportion of volume to surface than those of small dimensions, even though the proportions be chosen to favour a minimum of surface in both cases. It is therefore to be expected that large size gas engines will give greater economy than smaller ones. This has been fully realised in practice.

We have discussed very briefly the controlling factors determining the size of cylinder for a gas engine, and we now proceed to apply the conclusions arrived at. We shall base our calculated size upon the following data :—

1. Piston speed, 500 ft. per minute.
2. Compression before ignition, 80 lb. per square inch.
3. Stroke of engine, $1\frac{1}{2}$ times diameter of cylinder.

Let D = diameter of cylinder. Then stroke = 1·5 D Revolutions per minute

$$= \frac{\text{piston speed}}{2 \times \text{stroke}} = \frac{500}{3\text{D}}.$$

Explosions per minute (on Otto cycle)

$$= \frac{\text{revolutions}}{2} = \frac{500}{6\text{D}}.$$

Mean effective pressure may be taken approximately as

$$= 2\text{C} - 0{\cdot}01\,\text{C}^2 \text{ (when C = compression pressure)}$$
$$= 160 - 0{\cdot}01 \times 6400$$
$$= 96 \text{ lb. per square inch.}$$

The indicated horse power is to equal 25. Hence we have

$$25 = 1{\cdot}5\,\text{D} \times \frac{500}{6\text{D}} \times \frac{\text{D}^2\pi}{4} \times 96 \times \frac{1}{33000},$$

whence $$D^2 = 87{\cdot}5 \text{ (nearly)};$$

$$\therefore D = 9{\cdot}35 \text{ in., say } 9\tfrac{3}{8} \text{ in. diameter of cylinder.}$$

Stroke therefore equals $1{\cdot}5 \times 9{\cdot}35 = 14$ in., nearly.

Size of Combustion Chamber to give 80 lb. Compression (above Atmosphere).

Let V = whole volume,
= volume of combustion chamber + volume swept out by piston.
P = absolute pressure in pounds per square inch.

Then the following equation holds good for the compression curve—

$$P V^{1 \cdot 3} = \text{constant.}$$

To simplify the figures, the numerical value of V may be represented by linear inches, for, when the combustion chamber is the same diameter as the cylinder, the movement of the piston is proportional to the volume swept out. Hence the volume of the combustion chamber may be represented by

$$V - \text{stroke},$$
$$= V - 14.$$

At the commencement of the compression stroke the value of P will equal about 14 lb. per square inch absolute.

At the end of the compression stroke the pressure required is equal to 80 + 15 = 95 lb., absolute. We have therefore

$$P V^{1 \cdot 3} = P_1 V_1^{1 \cdot 3}$$

Taking P = 14 lb. per square inch, P_1 = 95 lb. per square inch, and $V_1 = (V - 14)$, we have

$$14 V^{1 \cdot 3} = 95 (V - 14)^{1 \cdot 3}$$

then $$V^{1 \cdot 3} = 6{\cdot}78 (V - 14)^{1 \cdot 3}$$

and $$V = \sqrt[1{\cdot}3]{= 6{\cdot}78 (V - 14)^{1 \cdot 3}}$$
$$= (V - 14) \sqrt[1{\cdot}3]{6{\cdot}78}$$
$$= (V - 14)\, 4{\cdot}36, \text{ nearly,}$$

from which $$3{\cdot}36 V = 61;$$
$$\therefore V = 18{\cdot}1.$$

Thus the actual volume of the combustion chamber, of any shape whatever, must

$$= (18{\cdot}1 - \text{stroke of piston}) \times \text{area of cylinder}$$

$$= (18{\cdot}1 - 14)\,\frac{D^2\,\pi}{4}$$

$$= 283 \text{ cubic inches.}$$

With respect to the arrangement of valves, it must be pointed out that the surface of all passages leading into the cylinder should be reduced as much as possible. The size of valves should be such that the velocity of the gases, as calculated upon the mean piston speed, is not more than 100 ft. per second. Although it is not possible to fully discuss all the details of cylinder design in the space at our disposal, it is hoped that we have sufficiently indicated how to determine the leading dimensions of a cylinder to develop a given horse power. By reference to the illustrations and descriptions of engines already given, there should be no difficulty in setting out the leading dimensions of an engine cylinder. It must not be forgotten, however, that the success of an engine depends upon minor details, a knowledge of which can only be acquired by practical experience in the working of gas engines. We make no attempt at describing the thousand and one minor details which will be readily supplied by the practical draughtsman, our object being rather to place before the reader those facts and figures which are not acquired in the erecting shop.

The Crank Shaft.

A gas-engine crank shaft should be made from the very best mild steel procurable. The severe stresses upon the shaft render it imperative that a large margin for strength be allowed. The author has known cases of gas-engine crank shafts of large diameter working successfully for many months, but which have suddenly failed, without apparent cause.

It is desirable that the crank webs be balanced by weight on the crank rather than on the flywheels. When the speed is high, and the flywheels of large diameter, balance weights fitted or cast on the wheels cause considerable oscillation of the whole engine. The bearings should be as close together as possible, and have ample surface.

To ascertain the true maximum twisting moment on a crank shaft when under normal conditions of working, it is necessary to correct the indicator diagram for the inertia of the reciprocating parts. This done, a polar diagram of twisting moment may be plotted for various positions of the crank. We have already stated that the maximum pressure upon the piston will be approximately = 3·5 times the compression pressure. Thus, in the example under consideration, we should expect a pressure of $80 \times 3{\cdot}5 =$ 280 lb. per square inch. This pressure occurs almost on the dead centre of the crank, and decreases as the tangential effort increases. We shall not be far wrong in taking 70 per cent of the maximum pressure as the load producing maximum twisting moment. Thus the maximum tangential effort may be taken as compression pressure $\times$ 2·5. When the bearings are close up to the crank webs, bending stresses may be neglected.

If D = diameter of shaft required, skin stress on the shaft 8,000 lb., then we have—

$$\frac{\pi}{16} D^3 \,.\, 8000 = 80 \times 2{\cdot}5 \times 7 \times 69\,;$$

$$D^3 = \frac{16 \times 80 \times 2{\cdot}5 \times 7 \times 69}{3{\cdot}14 \times 8000}\,;$$

from which $D = 3{\cdot}95$, say 4 in. diameter.

We are aware that this rule gives an exceptionally large diameter of shaft. We are, however, inclined to believe that the wisdom of an engine builder may be measured in terms of his crank-shaft dimensions, and we are therefore inclined to favour maximum dimensions.

The crank webs should be of ample size to give rigidity. The depth of the webs is, of course, largely determined by the diameter of the shaft. A good rule is to make the depth of web equal to

$$D + \frac{D}{3};$$

whilst the thickness of each web should be

$$= D - \frac{D}{8},$$

where D = diameter of shaft.

The crank pin should not be less in diameter than 1·2 D, the length being determined by the pressure upon the crank pin. In a gas engine (unlike a steam engine) the pressure upon the piston falls very rapidly; hence the maximum load per square inch may exceed that of steam-engine practice. The maximum pressure on a gas-engine crank pin should not exceed 1,000 lb. per square inch; but, with due regard to this, the load per square inch, as calculated upon the *average* piston pressure, may reach 400 lb.

Applying the rules given, our crank-shaft dimensions for the case under discussion will be as follow:—

Depth of web

$$= 3{\cdot}95 + \frac{3{\cdot}95}{3} = 5{\cdot}26, \text{ say } 5\tfrac{1}{4} \text{ in.}$$

Thickness of each web

$$= 3{\cdot}95 + \frac{3{\cdot}95}{8} = 3{\cdot}46, \text{ say } 3\tfrac{1}{2} \text{ in.}$$

The diameter of crank pin

$$= 3{\cdot}95 \times 1{\cdot}2 = 4{\cdot}74, \text{ say } 4\tfrac{3}{4} \text{ in.}$$

Then length of crank pin *(calculated on maximum pressure)*

$$= \frac{69 \times 280}{1000 \times 4{\cdot}75} = 4{\cdot}06.$$

Then length of crank pin *(calculated on average pressure)*

$$= \frac{69 \times 96}{400 \times 4{\cdot}75} = 3{\cdot}48.$$

Hence the length should be (say) $4\frac{1}{8}$ in., to satisfy the worst condition.

The main bearing pressures are largely affected by the weight of the flywheels, for the maximum pressure upon the bearings is the resultant of a thrust upon the crank and a vertically applied load due to the flywheels. A safe rule, which comprehends all conditions, is to allow 100 lb. per square inch, as calculated upon the mean piston pressure. Thus, in the example before us, we should have ample bearing surface by making the *total* lengh of bearings

$$= \frac{69 \times 96}{100 \times 4} = 16\tfrac{1}{2} \text{ in.}$$

Dimensions of Flywheels Required for Gas Engines.

In calculating the weight of gas-engine flywheels, we shall neglect the effect of inertia of reciprocating parts. Having regard to the fact that the wheels must drive for three strokes out of every four without undue fluctuation of speed, the effect of inertia becomes insignificant.

Let R_1 = maximum velocity, expressed as revolutions per minute;
R_2 = minimum velocity, expressed as revolutions per minute;
V_1 = maximum velocity in feet per second at the mean diameter of the wheels;
V_2 = minimum velocity in feet per second at the mean diameter of the wheels;
W = weight of flywheel in pounds.

Then mean revolutions per minute

$$= \frac{R_1 + R_2}{2}.$$

Average work done per stroke

$$= \frac{\text{H.P.} \times 33{,}000}{2 \times \frac{R_1 + R_2}{2}} = \frac{\text{H.P.} \times 33000}{R_1 + R_2}.$$

Let n = number of strokes the engine runs without impulse. (It may here be noted that no allowance need be made for the work done in compression during the idle strokes, for the energy thus absorbed is always given back again during the out strokes.)

The energy to be stored in flywheels

$$= \frac{\text{H.P.} \times 33000 \times n}{R_1 + R_2}.$$

Let the fluctuations in speed be represented as

$$\frac{V_2}{V_1} = C = \frac{R_2}{R_1};$$

then $$V_2 = C\,V_1,$$

and $$R_2 = C\,R_1.$$

Now, the energy absorbed during n revolutions, without impulse to the flywheels, must not reduce the speed below R_2 or V_2. Hence energy absorbed must equal

$$\frac{W}{2g}(V_1^2 - V_2^2) = \frac{W}{2g}(V_1^2 - C^2 V_1^2) = \frac{W V_1^2}{2g}(1 - C^2).$$

Let D = mean diameter of wheels, then

$$V_1^2 = \left(\frac{D \pi R_1}{60}\right)^2$$

Hence, by substituting this value of V_1^2, we have work done per n stroke = energy derived from flywheels—that is,

$$\frac{\text{H.P.} \times 33000 \times n}{R_1 + R_2} = \frac{W}{2g} \cdot \frac{D^2 \pi^2 R_1^2}{3600}(1 - C^2),$$

and putting $R_2 = C\,R_1$, we have—

$$W = \frac{\text{H.P.} \times 33000 \times n \times 2g \times 3600}{R_1(1 + C)\,D^2 \pi^2 R_1^2 (1 - C^2)},$$

and, reducing the weight to tons, we have—

$$W \text{ (in tons)} = 343900 \frac{\text{H.P.} \times n}{R_1^3 \times D^2 (1 - C^2)(1 + C)}.$$

We have here taken D as the mean diameter, instead of twice the radius of gyration; the error is, however, insignificant, and is more than compensated for by the absence of complications in the formula. The value of c for electric light work should be taken as 0·98, thus allowing a total variation of speed of 2 per cent. It is, of course, desirable to keep the flywheels as light as possible, not only on account of cost, but with the object of increasing the mechanical efficiency. For this latter reason, wheels of large diameter are to be preferred, inasmuch as the required weight varies inversely as the square of the diameter. Although not usual in gas-engine practice, it might be desirable to box in the arms of large wheels, to prevent losses due to air resistance. If flywheels are made too large, they become dangerous on account of centrifugal action on the rim of the wheel. For safety, the outside of the wheel should be limited to a speed of 100 ft. per second.

With respect to the value of n, we may say that it is undesirable to work a gas engine missing fire oftener than alternate cycles. This is indeed highly important with large engines, because small leakages of unburnt gas pass through the cylinder, and if not fired frequently, serious explosions are thereby liable to occur in the exhaust pipes. It is much safer, even though accompanied by loss in efficiency, to set the governor and gas valves to fire the engine every cycle at half load. We may take the value of n, therefore, as being equal to seven strokes.

Taking the following values for the symbols, we can find the weight of flywheels for a 25 indicated horse power engine.

$$\text{Weight in tons} = \frac{343900 \times 25 \times 7}{R_1{}^3 \times D^2 (1 - 0{\cdot}98^2)(1 + 0{\cdot}98)};$$

putting

$$D = 5 \text{ ft., and } R_1 = \frac{\text{piston speed}}{2 \text{ stroke}} = \frac{500}{2{\cdot}33} = 215,$$

we have—

$$W = \frac{343900 \times 25 \times 7}{215^3 \times 5^2 \times 0.04 \times 1.98} = 3.05 \text{ tons.}$$

Thus, if two flywheels be used, they should weigh a little more than $1\frac{1}{2}$ tons each. We have neglected the weight of the spokes and boss, but as the error favours steady running, there is no necessity for further complications.

Connecting Rods.—The determination of connecting-rod sizes for high-speed engines has already been fully dealt with in books on engine design. We may refer the reader to the chapter on connecting rods in Professor Unwin's treatise on machine design. Regarding gas engines as high-speed engines, we shall quote a formula from the above-mentioned work, which we think gives a result in agreement with practice. We are quite aware that many connecting rods will be found working satisfactorily which are nevertheless under the size given by the formula. Considering the complexity of the conditions, it is impossible, without devoting much space and re-writing much that has already been written, to deduce a formula embracing all conditions. The value obtained by the following rule may be excessive for slow-speed engines, but it may be regarded as safe for all cases.

Let d = mean diameter of connecting rod;
D = diameter of cylinder;
l = length of connecting rod;
p = initial pressure on the piston in lbs. per square inch;

then $$d = 0.038 \sqrt{\{D\, l \sqrt{p}\}}.$$

Having obtained a mean diameter, the rod may, if more convenient, be made of oval section. The mean diameter may be increased by about $\frac{1}{2}$ in. towards the crank-pin end of the rod, and diminished by the same amount at the piston end. The length of rod may be from five to six times the crank length.

For the example under consideration—

$$d = 0.038 \sqrt{\{9.35 \times 35 \sqrt{280}\}}$$
$$= 2.8 \text{ in.}$$

Hence the diameters may be, say, $2\frac{5}{8}$ in. and $3\frac{3}{8}$ in.

The Piston Pin.

The pressure on the piston pin may be much more than upon the crank pin, on account of the small relative movement of the rod at this end. A pressure of 600 lb. per square inch, as calculated upon the mean effective pressure, will give suitable proportions to the pin. The length of the pin may be about $1.4 \times$ diameter. It must here be noted that the design of any part should be carried out with due regard to the facilities in machining. For this reason it might be convenient to have the small end of the connecting rod the same width as the larger end, thus avoiding troublesome packings to set the rod level when machining.

The size of pin required will be as follows:—

Total mean pressure

$$= 69 \times 96 \text{ lb.}$$

Bearing area in square inches

$$= \frac{69 \times 96}{600} = 11.05.$$

Diameter of cylinder is $9\frac{3}{8}$ in. Allowing $5\frac{1}{2}$ in. for the total length of boss to receive the pin, we may take the length as

$$9\tfrac{3}{8} - 5\tfrac{1}{2} = 3\tfrac{7}{8}.$$

Then diameter of pin

$$= \frac{11.05}{3.87} = 2.6, \text{ say } 2\tfrac{5}{8} \text{ in. pin.}$$

The pin should be secured in the bosses of the trunk piston by set screws, to prevent rotation therein. A drip lubricator should be arranged to supply oil to the pin when working.

The piston should carry from three to six cast-iron packing rings, according to the pressure, of about $\frac{5}{8} \times \frac{1}{2}$ section.

The side shaft should be driven by screw gearing, the designs for which may be taken from any treatise upon gearing. The governor should be driven by bevel gearing from the side shaft, if requiring rotary motion.

We have already given data required for sizes of pipes, tanks, and fittings external to the engine itself. Let us conclude these few notes on gas-engine design with one word of warning. Never rely upon lock nuts *without* split pins to prevent them working completely off. We have known serious accidents to result from overlooking this apparently small detail. The vibration of all combustion engines renders it absolutely necessary that steady pins and positive lock nuts should be freely used in connecting the parts.

CHAPTER XVIII.

Producer Gas, and its Application to Gas Engines.

In the early days of the gas engine, one of the chief points in favour of its extended use was the convenience with which the ordinary coal gas of town supplies could be utilised as its motive power. So long as the power developed by gas engines was small, the commercial advantages resulting from their use would have been seriously minimised by the necessity of a large capital outlay. Thus, in the early days of the gas-engine industry, the motive power was entirely supplied from the ordinary town mains. Gas engines had been established as a commercial success for some four or five years before any other means of supplying them with motive power was recognised as practicable or economical. Nor was this state of things entirely due to lack of invention. It is indeed surprising how many names are associated with the development of a practical gas generator, suitable not only for gas engines, but for many

industrial processes, and especially for metallurgical purposes. In view of this, we shall merely explain the broad principles upon which all gas producers work ; but we shall refer specially to the work of Mr. Emerson Dowson, whose name has for many years been associated with the practical application of gas-producing plant for the supply of motive power to gas engines.

Any combustible substance when heated to a high temperature, either by the process of its own combustion or by the application of external heat, will be partially or wholly gasified. The most familiar example of this process is the ordinary household fireplace. The sudden blaze which so often results from stirring the fire is merely the burning of those gases which are rising continually through the red-hot fuel, but which are prevented from burning when passing through the hot fuel because of the inferior supply of oxygen. When, however, these gases mingle with the fresh air passing above the surface of the fuel, the oxygen renders them combustible. If the fire be dull, the heat at this point is insufficient to ignite the gases, and much of the gaseous fuel passes away unburnt. If, however, the fire be stirred to allow more air to enter its upper strata, then the gases are kindled by the heat, and the flame, thus started, rapidly spreads.

It will be seen from this that a simple gas producer might be made by charging a vessel with red-hot coke, then passing through it a stream of oxygen sufficient to form a combustible gas, but insufficient to allow complete combustion while passing through the red-hot fuel. This gas might be collected in a suitable reservoir, and burnt at will by the addition of a further supply of oxygen. The process of manufacture of the gas is represented symbolically by

$$CO_2 + C = 2\,CO.$$

The resulting gas, carbon-monoxide, has a calorific value of about 340 B.T.U. per cubic foot. For commercial consumption the oxygen is supplied to the hot coke or carbon

by passing air through the containing vessel. The resulting gas is therefore very largely diluted with nitrogen, which reduces its calorific value per cubic foot to about 112 B.T.U., or about one-sixth that of coal gas.

It is clear that, so long as atmospheric air alone is used as a vehicle for the oxygen necessary for combination with the carbon, a very inferior gas will be produced. In order to avoid this, another method has been resorted to, the result of which is the production of a gas known as water gas. The bare outline of the operation is as follows: Atmospheric air is passed through a chamber containing coke or anthracite until the whole mass of fuel is at a bright red heat. The gases formed during this process are not collected. The air supply is then entirely cut off, and superheated steam is passed through the incandescent fuel. This steam is decomposed into its elements, oxygen and hydrogen, and the process may continue until the temperature of the fuel falls to a point below which no further decomposition is possible. By this process as much as 50 per cent of free hydrogen is obtained, whilst the liberated oxygen re-combines with the carbon to form about 44 per cent of carbon-monoxide, together with traces of free oxygen, carbon-dioxide, and marsh gas. The calorific value of this gas is about 240 B.T.U. per cubic foot, and, moreover, the percentage of combustibles approaches 100. Hence this gas is eminently suited for gas-engine purposes, and would be largely used excepting for the intermittent action of the apparatus. In generators of this gas the production can only continue for periods of about 15 minutes. After this time steam must be turned off, and air again passed through the fuel, in order to raise it to the incandescent state. Some success has been attained by duplicating this apparatus, but the additional capital outlay required, and the introduction of simpler methods of equal efficiency as regards gas production, has prevented the general adoption of intermittent producers.

The method so successfully carried out by Mr. Emerson Dowson embraces a combination of the above-described

processes. From a specially-constructed nozzle a jet of steam is blown into the incandescent fuel. The steam induces a current of air to enter the fuel at the same time. The oxygen thus continuously supplied to the fire maintains it at a proper temperature, whilst the resulting gas is greatly enriched by the decomposition of the steam ; and, moreover, by the judicious regulation of the weight of air and steam passing through the fuel, the process is continuously carried on.

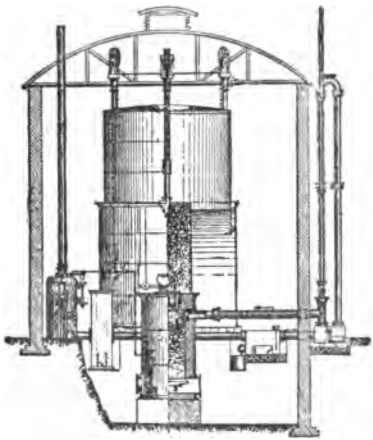


FIG. 98.—Dowson gas plant.

Figs. 98 and 99 show an elevation and plan of Dowson gas plant, of which the following is a brief description.

Referring to fig. 98, the boiler for generating the steam is shown at A. This boiler is kept at a pressure of from 30 lb. to 60 lb. per square inch, according to the size of the generator. Superheated steam passes over from the boiler through the pipe O to the gas generator shown at C. Beneath the firebars of the generator the steam passes through a nozzle of special construction, and so draws air with it into the closed ashpit

E. The gases resulting from the contact of the air and steam with the incandescent fuel in C pass up through the delivery pipes to the stand pipes F. The gases are here cooled, and are afterwards cleansed by passing through water in the box H. From H the gases pass to the scrubber J, and afterwards through the coke scrubber K placed inside the gasholder L, where they remain until conducted to the gas engine. The pressure in the gasholder is allowed to remain at about 1½ in. of water, and the pressure produced

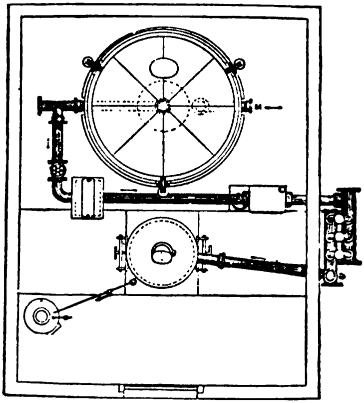


FIG. 99.—Plan of Dowson gas plant.

in the closed ashpit by the action of the steam jet is sufficient to maintain proper circulation of the gases from the producer to the gasholder. The supply of steam to the generator is regulated by a cock at O, the handle of which is attached to the upper moving portion of the gasholder L, thus automatically preventing the generation of gas when the holder becomes fully charged.

The fuel most suitable for consumption in the generator is anthracite, because of the absence of sulphur, smoke, and

other impurities. Its cost may be taken at from 12s. to 15s. per ton delivered, and the fuel required per 1,000 cubic feet of gas equals about 13·5 lb. Hence the cost of fuel in the generator per thousand cubic feet may be taken as about one penny. Including all charges, depreciation, wages, &c., upon a small plant capable of delivering 1,000 cubic feet per hour, the cost is worked out at 4¼d. per 1,000 cubic feet.* A larger plant, producing 3,000 cubic feet per hour, works out to about 2½d. per 1,000 cubic feet. It must here be remembered, however, that about five volumes of Dowson gas are required to evolve the same heat as one volume of coal gas. Hence the cost per 1,000 cubic feet must be multiplied by five before a comparison can be made between Dowson and coal gas. Thus we see that with large producers a saving of 67 per cent is effected, and with small producers a saving of 41 per cent, when coal gas is taken at 3s. per 1,000.

The lowest fuel consumption obtained with Dowson plant, working a twin-cylinder Crossley engine, indicating 118 horse power, was 0·76 lb. per indicated horse power hour during a working test of eight hours. When the losses in the generator due to clinkering and standing all night were added, the total consumption was 0·873 lb. per indicated horse power hour. The cost per indicated horse power hour, exclusive of wages and incidental expenses, is therefore 0·06d. This exceptionally low consumption of fuel can hardly be expected under ordinary working conditions. The average consumption may be taken as 1½ lb. of anthracite per indicated horse power hour.

Referring to analyses previously given of Dowson gas, it will be seen that it consists largely of carbon-monoxide, which gas is very poisonous. It is therefore important that all fittings should be tight, and that there should be good ventilation in the engine-room.

* Inst. Civil Engineers, Proceedings, vol. lxxiii.

It has already been mentioned that the calorific value of producer gas is much lower than that of coal gas; also that the air volume required for its combustion is less than for coal gas. Although some difficulty is found in practice in keeping the composition of the gas quite uniform, owing to choking up of the firebars, &c., yet an average composition of Dowson gas will be found to require, theoretically, 1·01 volumes of air to 1 of gas. In gas engines, 1·5 volumes of air to 1 of gas are usually adopted.

The following table, worked out similarly to that given previously for coal gas, will be found useful in working out the results of trials. For the purposes of these calculations, an average analysis is quoted:—

	Volume per cent.	Weight in one cubic foot of gas.	Proportion by weight.	Calorific value due to each constituent per cubic ft. at 32° and 14·7.	Weight of oxygen required per pound of gas.	Specific heat at constant volume.
Hydrogen	20	0·00111	0·008	58·27	0·064	0·01924
Carbon-monoxide..	23	0 01800	0·136	77·40	0·077	0·02352
Carbon-dioxide....	7	0·07420	0·561	....	....	0·09593
Nitrogen	50	0·03915	0·295	....	....	0·05103
	100	0·13246	1·00	135·67	0·141	0·18972

From the above table we see that 0·141 lb. of oxygen is required to combine with 1 lb. of Dowson gas. Hence

$$\frac{0·141}{0·23} = 0·613 \text{ lb.}$$

of air will be required to supply this weight of oxygen. 1 lb. of air occupies (at 32 deg. Fah. and 14·7 lb.) 0·08059 cubic feet. Volume of air required therefore

$$= \frac{0·613}{0·08059} = 7·61 \text{ cubic feet (nearly).}$$

1 lb. of Dowson gas occupies

$$\frac{1}{0.13246} = 7.56 \text{ cubic feet (nearly).}$$

The ratio of air to gas theoretically required is, therefore,

$$\frac{7.61}{7.56} = 1.01 \text{ (nearly).}$$

THE MOND GAS PRODUCER.

One of the greatest difficulties to be overcome in the design of a continuously working gas producer, is to prevent the formation of tarry deposit, which, if allowed to remain in the producer, soon renders it useless. With the object of preventing such a formation, it has been usual to burn only such fuel in the producer as is free from the tarry constituents, and consequently we find anthracite and coke generally used. With these fuels there are no by-products to be obtained in the process of gasification, whereas if a bituminous slack be used, a considerable quantity of ammonia might be recovered from the gas, the value of which may be credited to the producer, thus reducing the net cost of gas for purposes of motive power. It was with the object of bringing about this two-fold gain, namely the use of a cheaper bituminous slack and the ultimate recovery of the ammonia by-product, that Dr. Mond, of the firm of Messrs. Brunner, Mond and Company, Northwich, Cheshire, initiated experiments on a large scale, which have resulted in the design of the plant about to be described, and known as the Mond gas producer.* Cheap bituminous slack is fed by means of a creeper to a hopper H_1, over the producer. (Fig. 99A.) This hopper is connected at its base with a measuring hopper H_2. After the measuring hopper is filled the measured quantity of slack is permitted to pass into the producer through the counter-weighted hood-valve V. The producer P consists of a brick-lined cylindrical shell covered by a brick arch, through which depends a bell-shaped casting B opening by means of the hood-valve mentioned into the measuring hopper from which it receives its charge of

* Inst. C.E. Proceedings, vol. cxxix.

fuel. The lower end of the producer terminates in a coned neck containing radially inclined fire-bars F. The narrow end of the neck dips into a water-seal. It will be understood that the burning contents of the producer rest upon the

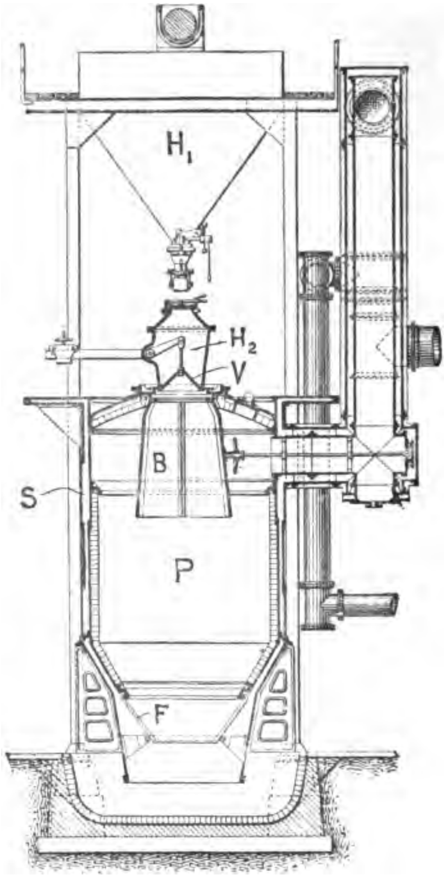


FIG 99A

inclined fire-bars and upon a conical column of ash standing upon the base of the water tank forming the water seal or lute. The ashes are withdrawn from beneath the surface of the water so that this may be done without interrupting the operation of the producer. A supply of steam and air is forced by means of a blower (see Fig. 100) under slight pressure, through a series of heating pipes R to be described, and so finds its way to the annular space S formed round the inner shell of the producer by the jacketing cylinder. By this arrangement the air and steam are heated as they pass to the producer by heat abstracted from the walls of the producer. The mixture of steam and air passes from the jacket, past the fire-bars and through the incandescent fuel, thus decomposing the steam and yielding the hydrogen constituent, at the same time forming carbonmonoxide. The gases from a fresh charge of slack are distilled while the latter remains in the inverted bell casting. The temperature of this casting is, of course, lower than that of the red hot fuel, and the success of the process seems to depend largely upon this fact. The tarry products of the distillation must pass through a much hotter zone than that in which they were distilled and are thus converted into fixed gases. These pass away together with the carbonmonoxide and hydrogen into a set of jacketed pipes R, in which the heat from the gases is given up to the air-and-steam mixture travelling towards the producer through the jackets. The gases next pass through a chamber C in which sprays of water are kept up by means of a mechanical dasher. From this chamber the gases pass to the bottom of the ammonia-recovering tower and are drawn from the top of this tower into a cooling tower. All the towers are filled with chequered brickwork. As the gas travels upwards through the ammonia recovery tower it meets a stream of acid liquor containing 4 per cent of free sulphuric acid. The ammonia is thus combined forming sulphate of ammonia. The cooling tower is served with water drawn from a cold water tank. The hot water resulting from the cooling process is delivered to

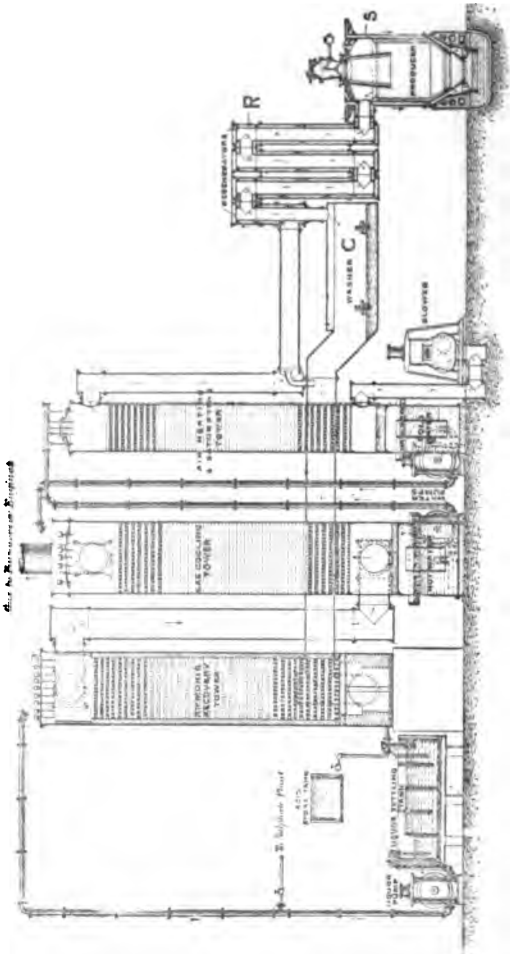


FIG. 100.

the air-heating tower. As the hot water meets the stream of cold air, the latter is heated and becomes saturated with water vapour and carries with it to the producer a considerable quantity of steam necessary for the formation of hydrogen. About one ton of steam per ton of fuel burnt in the producer is secured in this way, but a further 1½ tons is added from exhaust steam pipes or other sources.

This beautiful process of regeneration may be summarised as follows: The hot water from the cooling tower is used to impart heat to the air-and-steam mixture which is still further raised in temperature as it passes the vertical stack of pipes through which the hottest gases from the producer are travelling. It will be noticed that at the lower temperatures the transfer of heat is obtained by the agency of water spread over a very large surface on the checked brickwork, and at the higher temperatures the transfer takes place through the wall of the regenerator pipes.

The following are the analyses of various producer gases given by Mr. H. A. Humphrey:—

Volume per cent.	Mond producer gas from bituminous fuel.	Dowson producer gas from anthracite.	Lencauchez producer gas from anthracite.	Coal gas illuminating.	Solvay coke-oven gases.	Siemens producer gas.
Hydrogen (H)	24·8	18·73	20·0	48·0	56·9	8·6
Marsh gas (CH_4)	2·3	0·31	..	39·5	22·6	2·4
C_nH_{2n} gases	Nil.	0·31	4·9 (?)	3·8	3·0	Nil.
Carbonic Oxide (CO)	13·2	25·07	21·0	7·5	8·7	24·4
Nitrogen (N)	46·8	48·98	49·5	0·5	5·8	59·4
Carbonic Acid (CO_2)	12·9	6·57	5·0	Nil.	3·0	5·2
Total combustible gases	40·3	44·42	45·0	98·8	91·2	35·4
Theoretical Air required for complete combustion %	112·4	113·2	154·0	581·0	410·0	101·4
Calorific value per cub. ft. at 16° C., units at 18° C.	85·9	88·9	115·3	381·0	284·0	74·7
Ditto B.T.U.	155	160	208	685	512	135

It will be obvious that the capital outlay upon such a plant as that described is great, and in no case would it pay unless consuming upwards of twenty tons of coal per day. For an extensive system of electrical distribution, the Mond plant would be cheaply worked. In this connection the approximate price of a 10,000 horse power plant may be mentioned at £20,000. The comparative costs of working the Mond producer and the Dowson plant are given in Mr. Humphrey's paper as 0·507 pence per 1000 cubic feet of gas for the Mond plant, as against 1·8 pence for the same volume of gas from a Dowson plant. Although these figures may be accurate, they are not conclusively in favour of the Mond producer when regarded as a means of supplying fuel for gas engine purposes. The Mond producer which yielded this extraordinarily low figure of cost was considerably larger than any Dowson plant existing. We do not therefore look for any great change in the usual installations of medium size gas engine plants worked on Dowson gas. But it is undoubtedly true that the Mond producer, worked as it is on cheap bituminous slack, and affording means for the recovery of the ammonia by-product, will have a wide field of usefulness for extensive installations of 5,000 horse power and upwards.

THE LENCAUCHEZ GAS PRODUCER.

The Lencauchez producer is little known in England, but it is frequently applied to the Simplex engine in France where several such installations are working with poor French anthracite or non-bituminous coal. The producer is shown in fig. 101, and consists of a circular shell, inside which is a layer of sand backed with brickwork. Air is forced into a closed ash pit by means of a blower which may be driven by the engine worked on the producer gas. The air enters the closed chamber, passes through the fire-bars and hot fuel. A supply of water is delivered to the producer

* Reproduced by permission of the Council of the Institution of Civil Engineers.

through a pipe entering about twelve inches above the level of the grate bars. The evaporation of this water whilst dripping through the hot ashes, and whilst standing in the trough placed to receive the surplus of water below the bars, gives a sufficient quantity of steam to furnish about 20 per cent of hydrogen in the resulting gases. The products are washed in their passage through the coke scrubber, the coke in which is kept wet by a spray of water falling

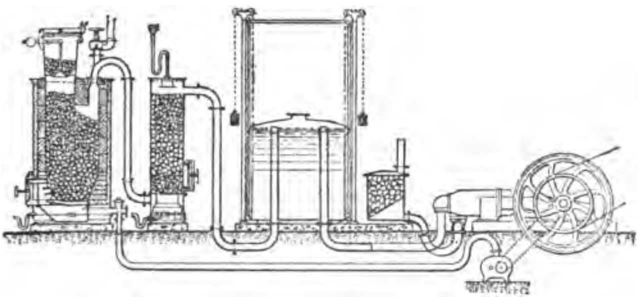


FIG. 101.

through it. From the scrubber the gases pass to the holder, and are ready for delivery to the engine. The small chamber shown at the back of the engine in fig. 101 is coupled to the exhaust pipe and acts as a silencer.

ON THE USE OF FURNACE GASES IN GAS ENGINES.

The utilisation of the waste gases from blast furnaces has occupied the attention of engineers for many years. It is only since the year 1895 that any attempts have been made to apply the gases to internal combustion engines, though a large proportion of the gases had previously been used for heating the blast and for steam raising purposes by burning the gases beneath steam boilers. An enquiry into the composition of blast furnace gases will help us to form an opinion as to how far they may be suitable for gas engine purposes. We have the authority of Sir I. Lowthian Bell,

that owing to the reducing condition which has to be maintained in the gases of an iron furnace, the proportion of carbon brought to the highest state of oxidation—that of carbonic acid—is limited to one-third of the whole. Thus it appears that the waste gases from such furnaces must, and always will, contain large proportions of carbonic oxide. It would appear, therefore, that no possible economy which may in the future be brought about in the working of blast furnaces will materially affect this element of the waste gases. This conclusion emphasises the importance of making every effort for the utilisation of iron furnace gases. The following analysis of waste gases from two furnaces are worked out in volumetric proportion.

ANALYSIS OF IRON FURNACE GASES.

Constituents.				Calorific value per cub. ft. at 14·7 lbs. sq. inch. 32 deg. F.	Remarks.
CO_2	CO	H	N		
12	27	1	60	98·9	Collected from furnace of 35013 cub. ft. on different days.
12·4	28·3	1	58·3	97·9	
9·5	29·2	2·9	58·4	106·5	From furnace of 20,454 cub. ft. on different days.
8·8	32·[illegible]	1·7	56·6	116	
Means. 10·68	29·35	1·65	58·32	103·5	Theoretical proportion of air required for combustion=103 per cent.

It has been urged that the great variation in the constitution of iron furnace gases prevents their use in gas engines. This contention has not been borne out by recent experiments, and, moreover, the analyses given of gases from two furnaces of different capacities refute the objection. It is, however, well to guard against variations by collecting the gases in suitable holders to permit of their diffusion.

When using coal gas, the mixture drawn into the cylinder of the engine is generally in the proportion of 10 volumes of air to one of gas. Thus, supposing the coal gas to have a

calorific value of say 640 units per cubic foot, in one cubic foot of mixture we should have 58 units of heat present. When Dowson gas is used the proportion is generally about $1\frac{1}{2}$ volumes of air to one of gas. Thus taking Dowson gas as having a calorific value of 150 units per cubic foot, we should have in a mixture of one cubic foot used in the cylinder 60 units of heat. In the case of iron furnace gases we see by the analysis that the proportion of air theoretically required is about the same as that for Dowson gas, namely 103 per cent of air. The excess of air required to form a satisfactory gas engine mixture is likely to be in about the same proportion as that required for Dowson gas. If this be true, a cubic foot of mixture of furnace gas would contain 41 units of heat. Thus it appears that in order to get equal powers developed by two engines, the one using Dowson gas, the other using furnace gas, the engine in the latter case would need to be of larger dimensions. The question arises, however, with so small a proportion of combustible gas, can a satisfactory ignition be obtained with certainty? This difficulty is met by increasing the initial compression of the mixture before ignition. Such compression effects two purposes. In the first place the higher compression brings the combustible gas into closer union with the particles of oxygen and thus facilitates the combustion; and in the second place the cooling surfaces of the cylinder exposed to the ignited mixture are diminished so that the temperature of the explosion more nearly approaches the theoretical temperature. In view of these facts, and the experimental work which we shall now briefly quote, there is no doubt that in the future great use will be made of the waste gases from, not only iron furnaces, but all furnaces yielding gases with a heating value of from 80 to 100 units of heat per cubic foot.

One of the practical difficulties which besets the use of furnace gases in gas engines is the presence of very fine metallic dust in the gases drawn from the furnace. In the experiments carried out by Mr. Bailly, Kraft, and Delamare

(the designer of the Simplex gas engine) at the works of the Société Cockerill, Seraing, near Liége, the gases were led from the furnaces through six coke scrubbers 5 ft. diameter and 19 ft. long. The gases were drawn from the furnaces by means of a Koerting steam jet, and to further assist the cleansing action of the steam jets about 6½ gallons of water per brake horse power was passed through the scrubbers. The water was afterward found to be unnecessary, the coke effecting all that was required. For testing purposes, the gases passed to a holder of 10,600 feet capacity; for the most part, however, the gases were taken direct to the engines. The gases in the engine were compressed before ignition to 115 lbs. per sq. in. Even after this treatment, there is said to be some dust left in the gases that reach the engine. In a 200 h.p. engine at Seraing there was said to be 88 lbs. of dust passing through the engine per day, but no inconvenience was caused, as the engine was run for a period of four months continuously without cleaning.

In 1898 a Simplex engine with single cylinder of 31½ inches diameter, and 3′–3½″ stroke, was tested under conditions above stated. The Indicated H.P. was 213, and the B.H.P. 181, thus giving a mechanical efficiency of 85 per cent. The calorific value of the gas was 109·8 B.T.U.'s per cubic foot, and 117·5 cubic feet were consumed in the engine per brake horse power hour. The speed of the engine was 105 revolutions per minute. This installation is now beyond the experimental stage, having run for more than 18 months without trouble in regard to dust or ignition.

The engines used on the Continent are larger and more powerful than those in general use in England. Experience on the Continent seems to point to the limit of power in one cylinder as being about 250 horse, at any rate where steady running is desirable. Engines of greater power are designed by duplicating the cylinders.

There seems to have been little done in England beyond the pioneer work by Mr. Thwaites, who initiated a scheme at the Glasgow Ironworks, Wishaw, in 1895. In addition to

this a gas engine plant has been successfully used at Barrow, and this is being extended.

The thermal efficiency of the engines tested with furnace gas is very encouraging, reaching as it does nearly 21 per cent. It is not likely that a steam plant supplied with the same fuel would yield more than from 7 to 8 per cent. thermal efficiency, or one-third that of the gas engine plant. In view of this extraordinary saving, there is a demand for a gas engine that will work with even a poorer gas than iron furnaces produce. For instance, one cannot but remark upon the great waste resulting from the use of coke ovens. It is true that the construction of coke ovens is undergoing improvement with a view of recovering the ammonia sulphate, and with a view to utilising the waste gases beneath boilers. It is, however, the ordinary, and much used, beehive type of oven that is the most wasteful. In a few instances the gases from this type of oven are used for steam raising, and there have been proposals to use gas engines, but from the writer's analyses of the gases from these ovens he does not think it likely that they will receive much attention in the direction indicated until iron furnace gases have been more widely exploited in the gas engine. The experience gained in building gas engines for iron furnaces will be of great use in adapting them to the still less heat-giving gas evolved from the coke oven.

The gases given off from a coke oven vary from day to day to a considerable extent. The charge of fuel is put into the oven, where it is destined to remain for from three to four days. After it has been in the oven about 8 hours gas may ignite at the crown of the oven as it passes out into the air. Towards the beginning of the second day the evolution of combustible gas is the greatest, and this gradually diminishes until the constituents of the gas are chiefly CO_2 and nitrogen.

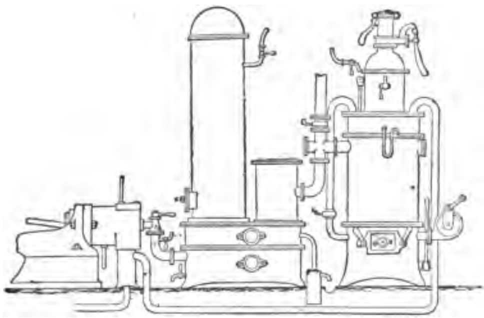
In coke oven gases the experience of the writer leads him to the conclusion that, at the best, the yield from one oven may be as high as 40 per cent. of combustible gas. This

gradually diminishes, and the average percentage of combustible gas during the charge will not be more than 20 The maximum heating value per cubic foot will be about 120 units, but the average will be about 60 units. Thus it appears that coke oven gas is of an inferior order to iron furnace gas. The only way in which this low calorific value can be met is, as we have already pointed out, by increasing the compression beyond that usually adopted. We have seen that 115 lbs. per square inch was adopted in the case of the iron furnaces at Seraing, and it is probable that in order to utilise the beehive coke oven gases an initial pressure of 130 lbs. per square inch would be needed to insure regular ignitions. Notwithstanding these possibilities, the convenience of the service of gas has to be carefully considered in relation to the labour at the disposal of the management. The manner in which coke ovens are used does not lend itself to accurate regulation of the flow of gas, and it is probable that the more careful attention needed would afford sufficient grounds for considerable prejudice, and will for some time act as a powerful deterrent from experiments in this direction. There is no doubt, however, that experiments with the beehive ovens are worth considering, and if an engine can be built—and there appears no reason why this should not be done—with the compression necessary for dealing with this gas, much economy in heat will be attained. Many managers will inform us that regularity and simplicity of working is worth more than heat efficiency, and whilst the former conditions are at stake, one cannot hope to do more than initiate small experimental plants.

THE DYNAMIC GAS PRODUCER.

This is a producer which is suitable for small power engines because of the very small floor space taken up. In the case of a 10 H.P. engine the approximate area occupied by the whole plant is only 9 square feet, and for a 60 H.P. engine about 30 square feet; the vertical height being in the first case 5 feet, and in the second 8 feet. The apparatus

consists of only two parts—the producer and the scrubber. The air required for combustion is heated by the exhaust gases from the engine, and the steam is produced at the expense of the heat in the hot gas as it leaves the producer, whilst the producer is also jacketed in order to utilize the radiant heat. The mixture of air and gas is drawn through the furnace into the scrubber, thence into a chamber, which is directly connected to the engine. The suction of the engine in drawing in the gas each stroke reduces the pressure in the apparatus slightly below that of the external atmosphere, and this suffices to draw the supply of air and gas through the incandescent mass of fuel, which is anthracite coal. The



The Dynamic Gas Producer.

fuel is supplied to the furnace by means of a hopper, two or three times per day, whilst the water required for the production of the steam can be automatically supplied from the mains by means of an ordinary ball-cock, because the pressure in the boiler is practically equal to that of the atmosphere.

The gas produced appears to be very clean and rich in heat-units, as in an experimental plant, the horse-power obtained is only about ten per cent less than with the town's gas, but at the time of going to press no exact tests have been made, so that the exact value of the gas produced has not yet been made. The Dynamic Company, however, show a saving in

the cost of working of from 50 to 60 per cent, according to the power developed by the engine when compared with the cost of town's gas, whilst its applicability to very small units of power, as well as large, and its low cost, and the total absence of all residuals, save a small quantity of ash (about 2 per cent), marks this invention as a most promising one for users of gas engines.

CHAPTER XIX.

The Effects of the Products of Combustion upon Explosive Mixtures of Coal Gas and Air.

Experiments previously made upon gaseous mixtures have been directed towards the investigation of the actual pressures produced by the combustion of an inflammable gas, in the presence of oxygen or pure air only. Thirty years ago such experiments were conducted in chemical laboratories on a very small scale quite incomparable with the volumes of the cylinders which are used in practical work. The practical difficulties which beset the development of the gas engine retarded, rather than stimulated, any very complete research upon the behaviour of explosive mixtures. Practical men were satisfied with an approximation to the maximum pressure which might be expected with any given mixture; this information was amply provided by Hirn, Bunsen, and later by Berthelot. As the mechanical arrangements became more efficient, the need was felt for further data regarding the comparative economy of various mixtures.

The most complete practical contribution upon this subject has been afforded by the experiments of Mr. Dugald Clerk, which enabled him to estimate the most economical mixture to be used in a non-compression engine, but no account was taken of the effects of the products of combustion which are present in the cylinder of a gas engine,

notwithstanding that the early engines were constructed with a clearance volume of 60 per cent. To obtain some definite data upon this important subject the author has recently carried out a series of experiments in the Engineering Laboratory of the Yorkshire College, Leeds.

It has been generally inferred that the products of combustion, when mixed with a fresh charge of coal gas and air, will decrease the maximum pressure and thereby reduce the efficiency of the charge; if this inference be correct, it would justify the efforts of gas-engine manufacturers in introducing a scavenger stroke, in order to minimise the evils of the presence of residual gases. The experiments carried out by the author show that the presence of the products of combustion in certain mixtures actually raise, rather than diminish, the maximum pressure obtained. It is, however, incontestible that the expulsion of the products of combustion has effected an increase in the economy of gas consumption; but, in the author's opinion, the reduced consumption per horse power is due to the increase in the effective cylinder capacity of any given engine when the products of combustion are replaced by explosive mixture, as well as to the cooling effects of the scavenging stroke. Following the example of previous experimenters upon gaseous mixtures, the author made use of a closed vessel of constant volume, and ignited the charges at atmospheric pressure. The use of such apparatus is considered by some to detract from the practical value of the results obtained, but if the presence of certain constituents affects the rise of pressure in a vessel of constant volume, it may be asked, Why should it not also affect the rise of pressure in the expanding chamber of a gas engine? The author is now continuing this series of experiments with compressed mixtures, the result of which he hopes shortly to publish.

The apparatus used for the explosion of the gases consisted of a thick cast-iron cylinder, flanged at both ends, of

1 cubic foot capacity. The cylinder (A, fig. 102) was bolted vertically to the column B.

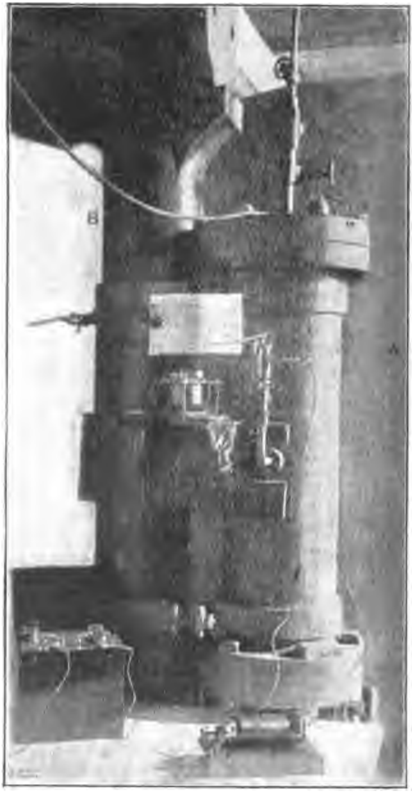


FIG. 102.

Igniting Arrangements. — The charge was ignited by passing an electric spark in the mixture by means of a secondary battery and induction coil, the wires from which passed through the insulated brass plug C on the top cover.

The insulation at first gave a great deal of trouble through moisture collecting on the under surface of the insulating material at the point where the wires penetrated (see fig. 103, *a*). The difficulty was ultimately overcome by adopting the arrangement shown in fig. 103, *b*. The brass casing, by projecting beyond the surface of the insulation,

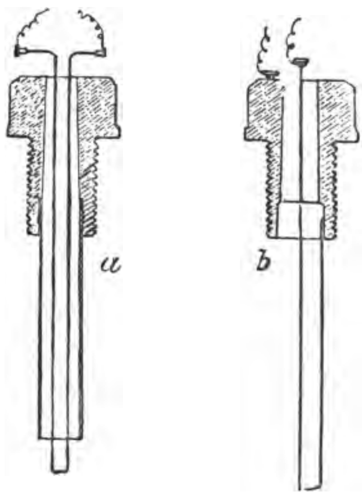


FIG. 103.—The Igniter.

prevented the water, during the filling of the cylinder, from rising to within the cup thus formed. By thoroughly drying, then oiling the surface of the insulation, the sparking was made certain. If any mixture failed to explode, the igniting gear was carefully tested, and the inflammatory nature of the gas ascertained by applying flame as it was gently driven from the experimental cylinder.

Temperatures.—The temperatures were measured, before ignition, by means of a thermometer enclosed in a wrought-iron case, containing mercury, which penetrated into the gas chamber.

Recording Gear.—The pressures were recorded by means of a Crosby indicator, the pencil of which was arranged to scribe upon a continuously-revolving drum, 8 in. diameter, driven by clockwork. The exact speed of this drum was checked by the vibrating spring, fig. 104, adjusted to make four complete oscillations per second, the dimensions of the spring being such that the inertia enabled it to overcome the slight friction of the pen during one experiment. By allowing this pen to remain stationary during one revolution of the drum, a zero line was traced, which was crossed at every eighth of a second by the wave line produced by the vibrating spring. This period of time is represented by a linear distance of 0·3 in., and thus further sub-divisions may be made if desired. It was inconvenient to arrange the pen of the time recorder immediately over

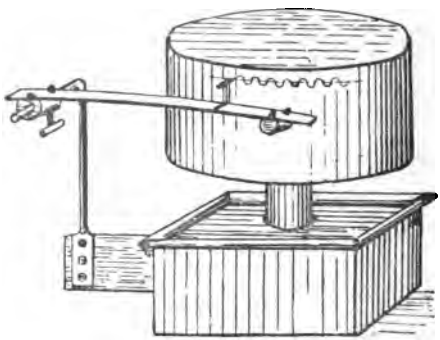


FIG. 104.—The Recorder.

the indicator pencil, but due correction has been made by transferring the required portion of the time wave to its proper position on the diagram.

In all the experiments the volumes were measured by filling the cylinder with water, and afterwards allowing the gas to enter as a measured quantity of water flowed out. After

firing the charge, a known volume of water was injected into the cylinder (care being taken that no air was inhaled by the partial vacuum formed by the condensation of the previous charge), thus expelling all but the required volume of residual gas ; this, together with a fresh supply of air, and the same volume of coal gas as before, formed the next charge. It need hardly be pointed out that the presence of products of combustion reduced the proportion of gas to pure air, for if this ratio had remained constant during the whole series, it would have necessitated a reduction in the quantity of coal gas, and consequently a reduction of the maximum explosion pressure. Undoubtedly the best arrangement would have been one in which the cylinder

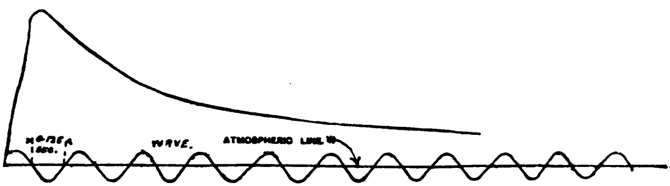


FIG. 105.—Indicator diagram, one volume coal gas, eight volumes air.

itself could have been subject to alteration in volume, for each experiment, sufficient to contain the quantity of products which might be present. The conclusions which may be deduced from these experiments involve the careful consideration of this point. The gases for each experiment were taken into the cylinder in the following order :—

1. The products of the previous combustion, if any.
2. Half the volume of pure air.
3. The coal gas.
4. The remaining air required to complete the charge.

No appreciable time was allowed for the diffusion of the mixture, it having been fired immediately after taking the temperature. It is impossible to estimate how the diffusion

of the gases in a working gas-engine cylinder is assisted by the rapid motion of the piston, though it is probable that the coal gas and air are more intimately mixed with one another than with the residual gases. In all experiments great care was taken to obtain atmospheric pressure in the cylinder, special precaution being taken to allow the excess of pressure of the coal gas to escape after uncoupling the supply.

The coal gas used throughout the experiments was taken from the service pipes of the Leeds gas supply. From analyses of three samples made by the author after the experiments, the composition was found to be as follows :—

ANALYSIS OF LEEDS COAL GAS.

Constituents.	Volume per cent.	Weight of 1 cubic foot.	Weight in 1 cubic foot of coal gas.	Proportion by weight.	Calorific value per pound.	Ditto, per pound of coal gas.	Proportion by weight of oxygen required for combustion.	Weight of oxygen required.
		Lbs.	Lbs.		B.T.U.			Lbs.
Marsh gas..........	35·2	0·0447	·01573	0·514	21,690	11,148	4	2·056
Olefines............	4·2	0·1174	·00493	0·161	20,260	3,261	$\frac{24}{7}$	0·552
Hydrogen..........	52·9	0·00559	·00295	0·096	52,500	5,040	8	0·768
Carbon monoxide...	6·5	0·0783	·00508	0·166	4,300	718	$\frac{4}{7}$	0·094
Nitrogen...........	0·1	0·0783	·00078	0·025	..	..	..	..
Carbon dioxide and oxygen...........	1·1	0·1060	·00116	0·038	..	..	..	..
	100·0	..	·03063	1·000	..	20,162	..	3·47

The mixtures experimented upon were as follow :—

Volume of coal gas.	Volumes of air.	Volume of coal gas.	Volumes of air.
*1	*16	1	12
1	15	1	10
1	14	1	8
1	13	1	6

* Failed repeatedly to explode.

Referring to Table I. (Appendix), it will be seen that after igniting a mixture of 1 volume of coal gas and 15 volumes of air, the residue was added to the next charge in the proportion of 5 per cent of the cylinder volume. Column 5 gives the actual pressure above the atmosphere as recorded by the indicator. In calculating the calorific values of the volumes of coal gas present in the experiments, the original temperatures and barometric pressures were allowed for; in other cases, the slight variations in initial temperature and barometric pressures have been neglected.

In Column VI. is recorded the rise of pressure above, or fall below, that obtained with a pure mixture, when the various percentages of residue were added. The greatest rise of pressure, namely, 19 lb. per square inch, took place when the products of combustion amounted to about 30 per cent of the whole cylinder volume, which left a proportion of coal gas to pure air of about 1 to 9½ by volume. The air was again decreased by the addition of residual gases until, in the proportion of 1 volume of gas to 6 of air, the rise in pressure was only 8 lb. per square inch above that recorded for the pure mixture. Fifty-five per cent of products was found to be the greatest quantity that might be introduced without preventing ignition. The results given in Table II. show that when the mixture was originally in the proportion of 1 volume of gas to 14 of air the explosions were somewhat less regular than in the previous cases, and

the maximum rise of pressure was greatest when the residual gases were present in the proportion of from 10 to 30 per cent, but only 12 lb. per square inch rise above the normal was recorded in this series.

The accompanying diagram, fig. 106, has been prepared from the figures in Column VI. of the tables. The ordinates above the line A B represent the excess of pressure above that recorded when mixtures of pure air and gas were exploded. Below the line the fall in pressure is plotted.

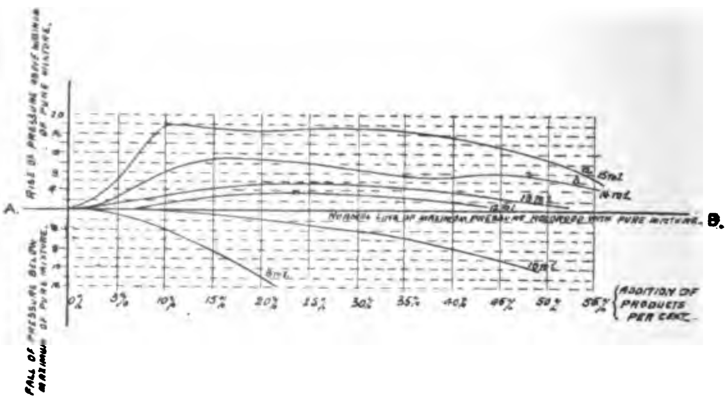


FIG. 106.—**Diagram showing rise and fall of maximum pressure due to the addition of products of combustion.**

The abscissæ represent the percentages of residue present. The curve marked *a* represents the rise of pressure with a mixture of 15 to 1. Similarly, the curve marked *b*, with a mixture of 14 to 1. When the volume of coal gas is as great as one-tenth of the cylinder volume, the addition of residual gases immediately reduces the pressure below that of a pure mixture. The general conclusions which may be drawn from this diagram are as follow: That with weak mixtures the maximum pressure is least when the excess of pure air is greatest. That the maximum pressure, obtained from a given quantity of coal gas, rises as the excess of air

is diminished by the addition of products of combustion, but this no longer holds when the volume of pure air to coal gas approaches the proportion of 10 to 1.

From an examination of fig. 107 it is seen that the explosion becomes more rapid as the excess of air is replaced by neutral gases. If the time is short compared with that of the outstroke of the motor, then the expansion line of an indicator diagram, taken under ordinary conditions, would be below the adiabatic curve. If, on the contrary, the burning continues during the whole of the outstroke, the expansion curve would probably be above the adiabatic, and would consequently give a greater mean effective pressure. Although in the diagrams before us we have only the curve due to cooling, we may still estimate the relative economy of a charge by measuring the area enclosed by the curve of pressure from its commencement to a point determined by an ordinate of time, which shall be chosen equal to that of the outstroke of a motor. Assuming the

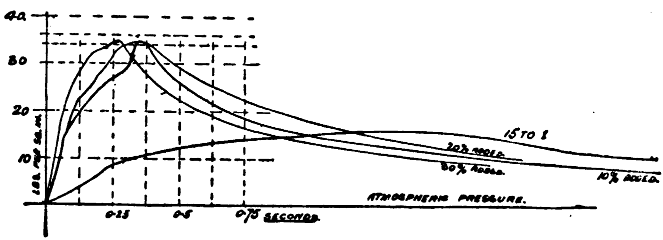


FIG. 107.—Diagram showing increase in the rate of combustion due to the addition of neutral gases.

rather slow speed of 0·25 second as being the duration of the outstroke, we obtain the following mean pressures derived from the areas of the pressure-time diagrams up to 0·25 second. It appears from the following calculations that the mean pressure is not influenced by the quantity of the products of combustion present, but it is always a maximum

when pure air is present in the proportion of about 10 to 1 of gas.

TABLE OF GREATEST MEAN PRESSURES, CALCULATED FROM PRESSURE-TIME DIAGRAMS, UP TO 0·25 SECONDS FROM COMMENCEMENT OF EXPLOSION.

Proportion of volume of coal gas to volume of cylinder.	Greatest mean pressure recorded. Pounds per square inch.	when— The percentage of products (estimated as clearance in an engine) was—	which gives— The proportion of pure air to coal gas, of—
$\frac{1}{16}$	9 20	None 30	15 to 1 11 to 1
$\frac{1}{15}$	11 23	None 20	14 to 1 11 to 1
$\frac{1}{14}$	11 21	None 18	13 to 1 11 to 1
$\frac{1}{13}$	10 21	None 18	12 to 1 10 to 1
$\frac{1}{12}$	32 33	None 17 8	11 to 1 8·3 to 1
$\frac{1}{9}$	42 40	None 11	8 to 1 7 to 1
$\frac{1}{7}$	30 26	None 5·2	6 to 1 5·6 to 1

From the composition of the coal gas used, it has been calculated that 5·7 volumes of air are required for complete combustion; and the experiments show that, as long as this ratio is not widely departed from, the mixture is explosive, notwithstanding the presence of 55 per cent of inert gases.

The results of all the experiments point conclusively to the fact that the ratio of air to gas alone determines the possibility of an explosion, and that the explosion pressure depends chiefly upon a suitable ratio being chosen.

The total heat evolved in each of the experiments was as follows:—

Heat values of the volumes of coal gas used.	Proportion of volume of gas to volume of cylinder.	Volume of air to volume of gas.
B.T.U.	Per cent.	
37	6·2	15 to 1
39	6·6	14 to 1
42	7·1	13 to 1
45	7·7	12 to 1
53	9·1	10 to 1
66	11·1	8 to 1
86	14·2	6 to 1

It was found, as might be anticipated, that the explosion pressures varied directly as the number of thermal units generated from the combustion of the gas. In fig. 108, the explosion pressures have been plotted as ordinates, and the number of British thermal units as abscissæ.

The diagrams in all cases show an alteration in the rate of combustion, in two places, but in no instance is there any sign of an actual reduction in pressure—merely an alteration in the rate of increase. Mr. Dugald Clerk, in his work upon the "Explosion of Gaseous Mixtures," produces an indicator diagram taken during the explosion of a mixture of one volume of gas to five volumes of air, in which the pressure curve distinctly falls after having reached a pressure of 60 lb. per square inch; it then rises to its maximum of nearly 100 lb. In none of the diagrams taken by the author, or in any single instance, out of some hundreds of experiments made by the Students of the College, has any such fall been recorded. It is therefore impossible to believe in a theory which attempts to explain it. Mr. Clerk's theory supposes that, after complete inflammation has taken place the pressure is further raised as the constituents of the coal gas combine with oxygen, only at a much slower rate than during inflammation. So far this may be satisfactory, but

to believe that this explains the momentary fall in pressure is, in the author's opinion, carrying the theory beyond its legitimate application.

It is found from an examination of the diagrams taken during the explosions of pure mixtures of air and coal gas, that the first alteration in the slope of the rising pressure curve occurs at 0·4 of the maximum height of the diagram. Whatever may be the real cause of this characteristic of all

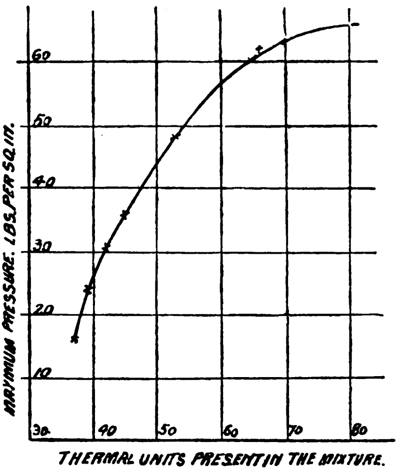


FIG. 108.—Diagram showing relation between maximum pressure obtained and thermal units present in the charge.

coal-gas explosion diagrams, it is interesting to note that the heat generated by the combustion of the hydrogen and the olefines combined is found to be just 0·4 of the total heat of the whole of the constituents of the coal gas experimented upon. Experiments have been made by Messrs. V. Meyer and F. Fryer, to determine the temperature at which each of the constituents of coal gas combine with oxygen. The results of their experiments are as follow :—

IGNITION TEMPERATURES OF EXPLOSIVE GASEOUS MIXTURES BURNT IN A CLOSED BULB.

Hydrogen combined with oxygen at	1,124 deg. Fah.
Olefines combined with oxygen at	1,124 deg. Fah.
Marsh gas combined with oxygen at	1,201 deg. Fah.
Carbon monoxide combined with oxygen at	1,347 deg. Fah.

It will here be noted that the hydrogen and olefines burn at the lowest temperature, namely, 1,124 deg. Fah., that the marsh gas burns next in order, and lastly the carbon monoxide at the highest temperature. From the calorific values of the constituents of the gas, which are given in a previous table, 1 lb. of coal gas is found to contain—

8,301 thermal units due to hydrogen and olefines.
11,148 thermal units due to marsh gas.
713 thermal units due to carbon monoxide.

20,162 total.

The total heat may, therefore, be divided into three parts, the proportion of each to the whole being—

0·417 of total heat due to hydrogen and olefines.
0·548 of total heat due to marsh gas.
0·035 of total heat due to carbon monoxide.

The annexed diagram (fig. 109) shows the application of these figures to the results. This diagram was taken during the explosion of 1 volume of gas mixed with 12 volumes of air. If horizontal lines are drawn through the points A, B, C, the figure is divided into three bands. If the specific heat be sensibly constant, then the pressures produced will be proportional to the heat developed at a given time. If, then, 0·41 of the total heat were developed first, we should expect to find the height of the first band to be 0·41 of the total height of the diagram. From several diagrams taken with pure mixtures the mean height of this band was found to be 0·403 of the total height. In the shaded diagram illustrating this point, the heights of the

bands are not quite in agreement with the previous figures, but very nearly so. The diagrams, taken when the residual gases are present in large volumes, show the point A is very much raised, probably on account of the hydrogen and olefines combining with the greater part of the oxygen present, the remainder being insufficient to completely burn the marsh gas and carbon monoxide. In other words, the total height of the diagram is not what it would have been had the heat due to the marsh gas and carbon monoxide been wholly developed.

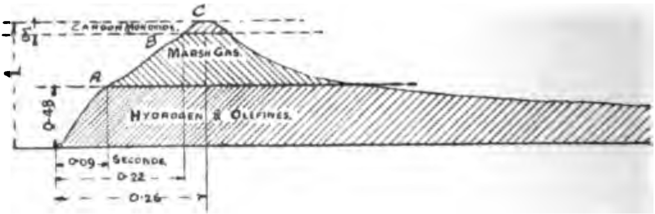


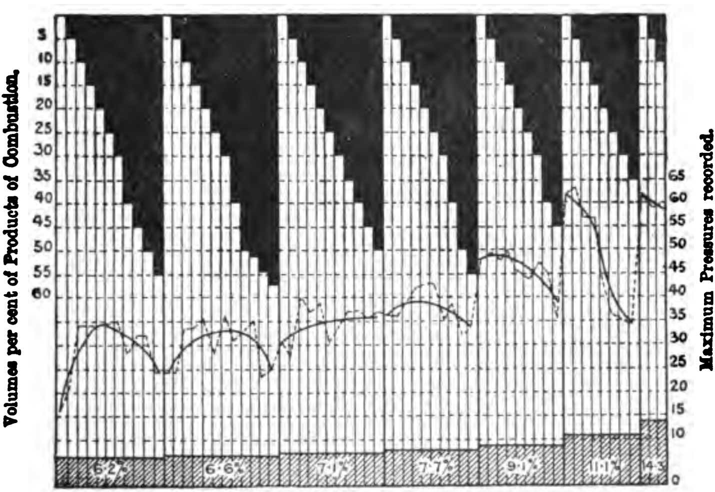
FIG. 109.

In every experiment the pressures recorded were low compared with those obtained by Clerk. Fig. 109 shows a maximum pressure of 38 lb. per square inch above the atmosphere, from which the maximum temperature works out to 1,350 deg. Fah. The maximum temperature possible was 3,250 deg. Fah., showing a loss of heat of 58·5 per cent, or 41·5 per cent accounted for, and the maximum pressure correspondingly higher. The difference is no doubt due to the fact that water was present on the walls of the cylinder, much of the heat thereby rendered latent in converting it into vapour. The pressure at the point A on fig. 6 is about 18 lb. per square inch above the atmosphere, and the heat accounted for at this point is found to be 49 per cent of that due to the hydrogen and olefines. That the heat accounted for at points on the pressure line near the atmospheric is more than at the maximum is what we should anticipate, for the cooling curves are all much steeper at the top than

lower, showing that the rate of transmission of heat is greatest at the highest temperature.

A summary of the whole serie of these experiments is shown by fig. 110. Plotted to a scale of 20 lb. = 1 in. are the pressures recorded during each experiment. The dotted line passes through all the points of pressure; the full line shows the mean rise and fall of pressure throughout.

Diagram showing Volumes of Coal Gas, Air, and Neutral Gases, with Maximum Pressures in pounds per square inch for each mixture.



Coal Gas Volumes, per cent.

FIG. 110.

CONCLUSIONS.

From these combined results the following conclusions may be drawn:—

1. That the highest pressures are obtained when the volume of air present is only slightly in excess of the amount required for complete combustion.

2. That higher pressures are recorded when residual gases take the place of an *excess* of air.

3. That when the volume of the products of combustion does not exceed 58 per cent of the mixture, then it is explosive, provided the volume of air is not less than 5·5 times the volume of coal gas.

4. That the time of an explosion is much reduced when excess of air is replaced by products of combustion.

APPENDICES TO PART I.

Tables of Results of Experiments on the Effects of the Products of Combustion upon Explosive Mixtures of Coal Gas and Air. (Referred to in Chapter XIX.)

Table I.

Col. I.		Col. II.		Col. III.		Col. IV.		Col. V.	Col. VI.
Ratio of volume of coal gas to volume of cylinder.		Ratio of volume of coal gas to volume of air.		Ratio of volume of products to volume of cylinder.		Ratio of volume of products to volume of air+coal gas.		Maximum pressures, pounds per square inch.	Rise of pressure above (+) or fall below (−) maximum for pure mixture.
Actu'l	Per cent	Actual	Per cent	Actu'l	Per cent.	Actu'l	Per cent.		
1/16	6·2	1/15	6·6	0	0	0	0	16	..
,,	,,	1/14·2	7·0	1/20	5	1/19	5·2	22	+ 6
,,	,,	1/13·2	7·4	1/10	10	1/9	11·1	34	+ 18
,,	,,	1/12·6	7·9	1/6·6	15	1/5·6	17·8	34	+ 18
,,	,,	1/11·8	8·4	1/5	20	1/4	25	34	+ 18
,,	,,	1/11	9·0	1/4	25	1/3·3	30	34	+ 18
,,	,,	1/10·2	9·8	1/3·3	30	1/2·3	45	35	+ 19
,,	,,	1/8·6	11·6	1/2·5	40	1/1·5	66	28	+ 12
,,	,,	1/7·8	12·8	1/2·2	45	1/1·2	89	32	+ 16
,,	,,	1/7	14·3	1/2	50	1/1	100	32	+ 16
,,	,,	1/6·1	16·2	1/1·8	55	..	..	24	+ 8
,,	,,	1/5·3	18·6	1/1·6	60	..	..	Failed.	..

Table II.

1/15	6·6	1/14	7·1	0	0	0	0	24	..
,,	,,	1/13·3	7·5	1/20	5	1/19	5·2	24	0
,,	,,	1/12·5	8	1/10	10	1/9	11·1	33	+ 9
,,	,,	1/11·8	8·4	1/6·6	15	1/5·6	17·8	34	+ 10
,,	,,	1/11	9·1	1/5·0	20	1/4	25	36	+ 12
,,	,,	1/9·7	10·3	1/4	25	1/3·3	30	28	+ 4
,,	,,	1/9·5	10·4	1/3·3	30	1/2·3	45	36	+ 12
,,	,,	1/8	12·4	1/2·5	40	1/1·5	66	31	+ 7
,,	,,	1/6·6	15·1	1/2	50	1·1	100	33	+ 9
,,	,,	1/6·2	16·1	1/1·9	52	..	..	35	+ 11
,,	,,	1/5·7	17·3	1/1·8	55	..	..	23	− 1
,,	,,	1/5·2	19·2	1/1·7	58	..	..	25	+ 1
,,	,,	1/5	19·9	1/1·6	60	..	.	Failed.	

TABLE III.

Col. I.		Col. II.		Col. III.		Col. IV.		Col. V.	Vol. VI.
Ratio of volume of coal gas to volume of cylinder.		Ratio of volume of coal gas to volume of air.		Ratio of volume of products to volume of cylinder.		Ratio of volume of products to volume of air+coal gas.		Maximum pressures, pounds per square inch.	Rise of pressure above (+) or fall below (−) maximum for pure mixture.
Actu'l	Per cent	Actual	Per cent	Actu'l	Per cent	Actu'l	Per cent		
1/14	7·1	1/13	7·7	0	0	0	0	31	..
,,	,,	1/12·3	8·1	1/20	5	1/19	5·2	27	− 4
,,	,,	1/11·6	8 6	1/10	10	1/9	11·1	40	+ 9
,,	,,	1/11	9·1	1/6·6	15	1/5·6	17·8	37	+ 6
,,	,,	1/10·2	9·8	1/5	20	1/4	25	39	+ 8
,,	,,	1/9·5	10·5	1/4	25	1/3·3	30	30	− 1
,,	,,	1/8·8	11·3	1/3·3	30	1/2·3	45	34	+ 3
,,	,,	1/8·1	12·3	1/2·8	35	1/1·8	54	37	+ 6
,,	,,	1/7·4	13·5	1/2·5	40	1/1·5	66	37	+ 6
,,	,,	1/6·7	14·9	1/2·2	45	1/1·2	89	36	+ 5
,,	,,	1/6	16·9	1/2	50	1/1	100	37	+ 6
,,	,,	1/5·3	18·9	1/1·8	55	1/	..	Failed.	..

TABLE IV.

1/13	7·7	1/12	8·3	0	0	0	0	36	..
,,	,,	1/11·3	8·8	1/20	5	1/19	5·2	36	0
,,	,,	1/10 7	9·3	1/10	10	1/9	11·1	40	+ 4
,,	,,	1/10	9·9	1/6·6	15	1/5·6	17·8	42	+ 6
,,	,,	1/9·4	10·6	1/5	20	1/4	25	41	+ 5
,,	,,	1/8·7	11·4	1/4	25	1/3 3	30	43	+ 7
,,	,,	1/8·1	12·3	1/3·3	30	1/2·3	45	35	− 1
,,	,,	1/6 8	14·7	1/2·5	40	1/1·5	66·7	39	+ 3
,,	,,	1/5·5	18·3	1/2	50	1/1	100	32	− 4
,,	,,	1/4·8	20·8	1/1·8	55	..	..	34	− 2
,,	,,	1/4·4	22·7	1/1 7	58	..	..	Failed.	..
,,	,,	1/4·1	24·3	1/1 6	60	..	..	Failed.	..

Table V.

Col. I.		Col. II.		Col. III.		Col. IV.		Col. V.	Col. VI.
Ratio of volume of coal gas to volume of cylinder.		Ratio of volume of coal gas to volume of air.		Ratio of volume of products to volume of cylinder.		Ratio of volume of products to volume of air+coal gas.		Maximum pressures, pounds per square inch.	Rise of pressure above (+) or fall below (−) maximum for pure mixture.
Actu'l	Per cent	Actual	Per cent	Actu'l	Per cent	Actu'l	Per cent		
1/11	9·1	1/10	10	0	0	0	0	48	..
,,	,,	1/9·4	10·6	1/20	5	1/19	5·2	49	+ 1
,,	,,	1/8·9	11·2	1/10	10	1/9	11·1	48	0
,,	,,	1/8·3	12	1/6·6	15	1/5·6	17·8	50	+ 2
,,	,,	1/7·8	12·8	1/5	20	1/4	25	45	− 3
,,	,,	1/7·2	13·8	1/4	25	1/3·3	30	44	− 4
,,	,,	1/6·7	14·9	1/3·3	30	1/2·3	45	47	− 1
,,	,,	1/5·6	17·9	1/2·5	40	1/1·5	66·7	45	− 3
,,	,,	1/5	19·8	1/2·2	45	1/1·2	82	35	− 13
,,	,,	1/4·9	20·2	1/2·1	46	1/1·7	85	0	..
,,	,,	1/4·7	21·3	1/2·08	48	1/1·1	92	0	..
,,	,,	1/4·5	22·2	1/2	50	1/1	100	0	..

Table VI.

1/9	11·1	1/8	12·5	0	0	0	0	62	..
,,	,,	1/7·5	13·4	1/20	5	1/19	5·2	63	+ 1
,,	,,	1/7·1	14·2	1/10	10	1/9	11·1	57	− 5
,,	,,	1/6·6	15 2	1/6·6	15	1/5·6	17 8	57	− 5
,,	,,	1/6·2	16·1	1/5	20	1/4	25	43	− 19
,,	,,	1/5·7	17·4	1/4	25	1/3 3	30	36	− 26
,,	,,	1/5·3	18·8	1/3·3	30	1/2·3	45	35	− 27
,,	,,	1/4·8	20·6	1/2·8	35	1/1·8	54	35	− 27
,,	,,	1/4·6	21·8	1/2·6	38	1/1·6	61	Failed	..

Table VII.

1/7	14·3	1/6	16·6	0	0	0	0	62	..
,,	,,	1/5·8	17·7	1/20	5	1/19	5 2	59	− 3
,,	,,	1/5 3	18·6	1/10	10	1/9	11·1	60	− 2
,,	,,	1/4·9	20·2	1/6·6	15	1/5 6	17·8	Failed	..

RESULTS OF GAS-ENGINE TRIALS. TOWN GAS USED.

Name of engine.	I.H.P.	B.H.P.	Mechanical efficiency.	Cubic feet of gas per I.H.P. hour.	Cubic feet of gas per B.H.P. hour.	Authority and remarks.
Otto-Crossley	17·1	14·7	0·86	20·7	24·1	Society of Arts Report.
Clerk's engine	9·05	7·23	0·8	24·3	30·4	Clerk's "Gas Engines," page 141.
Atkinson's Differential	..	2·6	..	..	25·7	Robinson's "Gas and petroleum engines."
Atkinson's Cycle	5·56	4·89	0·88	19·78	22·5	Professor Unwin.
Griffin	15·47	12·51	0·8	23·1	28·5	Professors Kennedy and Jamieson.
Forward engine	5·54	4·8	0·86	20·79	23·97	Professor Smith.
Simplex	..	8·79	..	..	20·38	Witz.
Wells Brothers small gas engine	8·5	11·9	0·71	20	27·8	At half load. Professor Goodman.
Wells Brothers small gas engine	13·5	17·6	0·76	16·3	21·2	Nearly full load. Professor Goodman.
Ditto ditto Premier Cycle	47	61	·77	15·27		Full load. Professor Goodman.

TRIALS IN WHICH DOWSON GAS WAS USED.

Name of engine.	I.H.P.	B.H.P.	Anthracite consumed in producer per I.H.P. Pounds.	Ditto B.H.P. Pounds.	Authority.
Crossley Otto	27·5	..	1·4	.	Inst. Civil Engineers. Proc., vol. lxxiii.
Crossley Otto	118·7	..	0·762	..	The *Engineer*, February 12, 1892.
Atkinson's Cycle engine	21·9	..	1·06	..	The *Engineer*, February 12, 1892.
Simplex	110	75·86	..	1·31	
Stockport	76	..	0·86	..	Mr. H. Robb.

*TABLE SHOWING THEORETICAL INDICATED EFFICIENCY OF CROSSLEY OTTO ENGINES WITH DIFFERENT COMPRESSIONS, COMPARED WITH ACTUAL INDICATED EFFICIENCIES WITH THE SAME COMPRESSIONS.

Calculated efficiency for perfect Otto cycle engine from compression space volume.	Actual indicated efficiency from diagram and gas consumption.	Ratio of actual to ideal efficiency.	Cylinder diameter. in inches	Piston stroke in inches.	Ratio of compression space to space swept by piston.	Pressure of compression in pounds.	Gas consumption per I.H.P. per hour in cubic feet.
0·33	0·15	$\frac{0·15}{0·33} = 0·45$	9	13	0·6	38·0	24·0
0·4	0·189	$\frac{0·189}{0·40} = 0·47$	$9\frac{1}{2}$	18	0·4	61·6	20·5
0·393	0·19	$\frac{0·19}{0·393} = 0·48$	$8\frac{1}{2}$	18	0·418	53·0	20·3
0·428	0·25	$\frac{0·25}{0·428} = 0·58$	7	15	0·34	87·5	14·8

*TABLE SHOWING COMPARISON OF THE ACTUAL AND THEORETICAL EFFICIENCIES OF OTTO ENGINES OF DIFFERENT DIMENSIONS.

Engine cylinder.		Relative capacity.	Theoretical efficiency.	Actual indicated efficiency	Rates of actual and ideal efficiency.
Nearly equal compression ..	7 in. diameter by 15 in. stroke	1 00	0·428	0·250	$\frac{0·25}{0·428} = 0·58$
	$11\frac{1}{2}$ in. diameter by 15 in. stroke	3·77	0·428	0·275	$\frac{0·275}{0·428} = 0·64$
	$9\frac{1}{2}$ in. diameter by 18 in. stroke......	1·00	0·40	0·210	$\frac{0·21}{0·41} = 0·53$
	14 in. diameter by 25 in. stroke......	2·97	0·41	0·277	$\frac{0·277}{0·41} = 0·67$

*Clerk on "Gas Engines." Proc. Inst. C E.

CHAPTER XX.

ACETYLENE GAS.

Acetylene gas is now used for lighting purposes in isolated towns and villages. With the object of ascertaining how far this gas is applicable to gas engines, the following experiments have been made. The apparatus used was a modification of that already described. A gas holder was added by which the volume of acetylene could be accurately measured without introducing water into the explosion cylinder. The

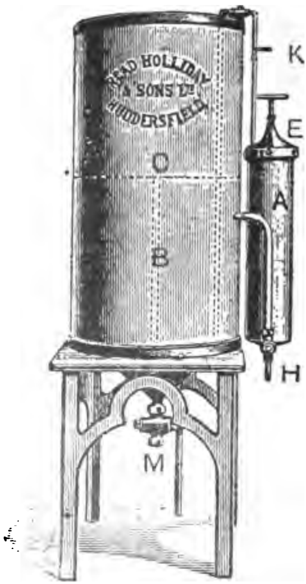


FIG. 111.

igniting arrangement and the manner in which the times were recorded were alike improved upon.

The experiments were carried out in the following way:—A known body of acetylene gas was admitted to a cylinder, and time allowed for its diffusion with the air therein. The mixture was ignited by electricity, and the pressure developed was measured by means of a Crosby indicator, the pencil of which worked upon a drum revolving at a known speed. In this way the proportions of acetylene and air, the time taken to complete the inflammation, and the pressures developed were observed. The products of combustion were analysed and the original mixtures checked. When any discrepancy was found, the quality of the original mixture was determined from the analysis of its products.

The acetylene gas was generated from an apparatus, illustrated in Fig. 111. The calcium carbide was placed in the part A. The parts C B were filled with water up to about six inches from the top. After taking precautions to free the generator from air, the acetylene was drawn off through a pressure-regulating valve attached to the pipe K. Reference will be made later to the purity of the acetylene obtained.

The volumes of gas were measured by means of an apparatus shown at Fig. 112. The vessel C consists of a copper pipe of uniform bore, closed at the top, and fitted with a pipe P for the transmission of the gas. The open end of this vessel is immersed in water contained in W. An open glass tube T is fitted to the lower end of W, so that the height of the water may at any time be observed. The tube T slides in a graduated tube, rigidly attached to C. The water vessel W is supported by balance weights, which permit of it being raised to displace a measured volume of gas into the explosion cylinder E. Any error due to the displacement of the water by the vessel C is eliminated; and by taking measurements at atmospheric pressure, the volume of gas used is accurately determined.

The explosion cylinder is shown at E, and consists of a

cast-iron pipe flanged at both ends. The cock A was used for admitting the acetylene from the meter.

The ignitions were effected by means of an internal contact-breaker, I, coupled in series with a glow lamp to the

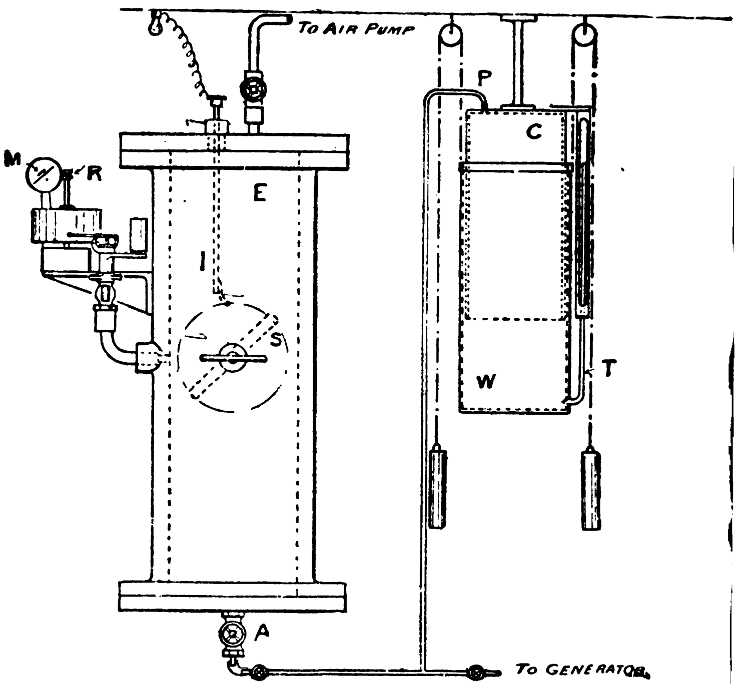


Fig. 112.

electric light leads of the building. The glow of the lamp sufficed to show that a contact had been made; and by quickly switching off the current, the spark necessary for igniting the mixtures was produced. The rotating wing S, which constituted part of the igniter, was also used for

stirring the mixture before ignition, to assist the diffusion. When this was being rotated for this purpose, it was temporarily disconnected from the electric light leads so that no sparks were then generated. This sparker has several advantages for experimental work, and has been adopted in these experiments in preference to the arrangement by which a high tension current sparks across an air gap. The advantages are, that the contact points are kept clean by friction; that there is no trouble with defective insulation and consequent short circuiting of a high tension current; that moisture does not affect the sparker; and lastly that the Experimenter has evidence of the efficiency of the apparatus by the glow of the lamp and its subsequent extinction. This is extremely important, because the failure of a mixture to explode may be due to the inefficiency of the sparker, and when this is so, it cannot, with this arrangement, be falsely ascribed to the mixture.

A Crosby indicator was used for determining the pressure. The pencil was held by a light spring to the revolving drum (fig. 112), driven by clockwork. To measure the time from the commencement of ignition to the completion of inflammation, it was necessary that the drum should rotate quickly. The peripheral speed of the drum was 38·4 inches per second, so that one-hundredth of a second was represented by 0·384 of an inch measured horizontally on the diagram drawn by the indicator. The speed of the drum was obtained in the following way. The watch, M (fig. 112), was mounted vertically on a small spur wheel geared to the worm R, rotating with the drum. The worm rotated the watch contra-clockwise. Hence for a certain speed of the drum the centre-second hand became stationary. On the centre of the hand a small mirror was fixed, so that by observing the image of any distant object in the mirror, it was easy to determine accurately when the hand was stationary. At this instant the gases were ignited by turning the wing S; the result was that the surface speed of the paper was always the same at the instant of ignition.

This method of securing a uniform time record is an extremely accurate one, and may be performed with comparatively rough apparatus; and moreover the principle is one which may be applied with modifications to measure time in a variety of laboratory experiments.

For compressing the mixtures before ignition, a Westinghouse air pump was used.

The first series of experiments consisted in exploding mixtures of acetylene and air at atmospheric pressure. The temperature before ignition was observed after the gases had diffused for 10 minutes. Mixtures ranging from 18 volumes of air to 1 of gas, and 4 of air to 1 of gas were exploded. No weaker mixture than 18 to 1 could be fired at atmospheric pressure. The superior limit was not obtained by experiment, but it is known that mixtures of 1½ of air to 1 of gas will explode at atmospheric pressure, and that by heating pure acetylene when compressed to two atmospheres it explodes without air. The pressures obtained are given in the tables appended, from which the points on Fig. 113 have been plotted. On the same diagram the corresponding pressures obtained with mixtures of coal-gas and air exploded in the same apparatus are plotted. It will be noticed that with weak mixtures of acetylene and air, the pressure is more than three times as great as with the same mixtures of coal-gas and air. But with stronger mixtures of acetylene the increase of pressure is less than twice as great. In making such comparisons of the two gases it must be remembered that coal-gas requires 5.7 volumes of air to 1 of gas, whereas acetylene requires 12.5 volumes of air to convey the necessary oxygen for its complete combustion. This fact accounts entirely for the relative positions of the curves on the diagram.

Not less remarkable than the increase of pressure, is the reduction of time for the complete inflammation of the gases. Thus it was found that inflammation was complete with acetylene mixtures in from 0·1 to 0·018 of a second, whereas with coal gas the times observed for the same mixtures were 0·5 to 0·25 of a second. 15 to 1 is the weakest

mixture of coal gas that can be exploded at atmospheric pressure, but with acetylene the limit is 18 to 1. The maximum pressure recorded was with a mixture of 7 to 1. Subsequent experiments with the mixtures at more than one atmosphere, showed that the true mixture to give a maximum pressure is nearer 11 to 1. The regularity of the points plotted on Fig. 113, indicate that the mixtures of air and gas were correctly measured. Subsequent experiments with the mixtures fired at more than one atmosphere gave some-

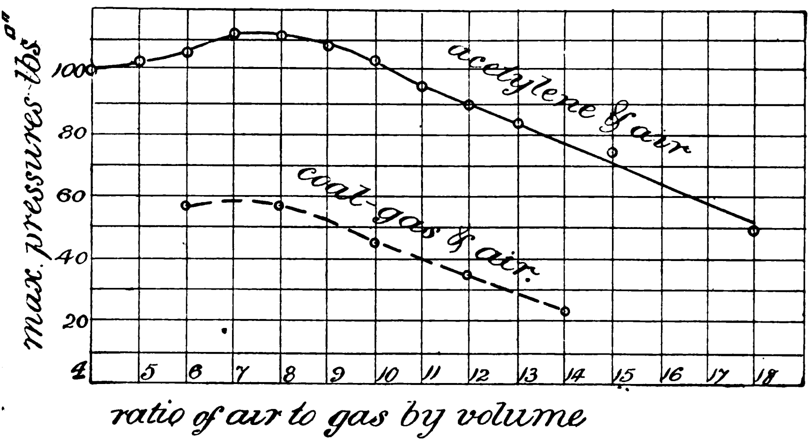


FIG. 113.

what discordant results, and lead the author to suspect that the true proportions of air to gas were not always identical with the proportions as measured by the apparatus. In the last series of experiments, where the discrepancies were most marked, the products of each combustion were analysed, and it was found that the true mixtures calculated from the analysis of the products of combustion were weaker than the supposed mixtures. It was at first thought that the escape of the gases from the cylinder during the time

allowed for their diffusion at the higher initial pressures was the cause of this discrepancy. That this was not the sole cause was shown by an analysis of gas from the holder after it had been drawn from the acetylene generator. The acetylene which was supposed to be pure was found to contain from 6 per cent. to 20 per cent. of incombustible gas—chiefly air. The greater proportion of air was found in the generator immediately after charging it; but as the gas was drawn off the proportion of air gradually diminished. In no instance is it likely that pure acetylene gas is delivered from a small generator because of the inevitable secretion of air and the diffusion of the gas with it during its expulsion. This is not detrimental when the generator is used for lighting purposes, but care should be exercised in order to avoid ignition of an explosive mixture just after charging. The errors possible in the mixtures in the explosion cylinder due to the air in the generator may be estimated as follows:—

Let x equal supposed volume of air in explosion cylinder when the volume of gas is supposed to equal 1. Then the supposed ratio of air to gas is represented by $x/1$.

Let $1/y$ equal actual volume of air in gas holder per 1 volume of gas and air.

Then the true ratio of air to gas in the explosion cylinder equals

$$\frac{x+1/y}{1-1/y} \text{ which reduces to } \frac{xy+1}{y-1} \text{ equals } \frac{\text{true volume of air.}}{\text{true volume of gas.}}$$

Now when $1/y=1/5=20\%$ of air in generator, true volume of gas to air in explosion cylinder becomes $1{\cdot}25x+0{\cdot}25$.

Now when	$1/y=1/10=10\%$	$1{\cdot}1x \quad +0{\cdot}1$
„ „	$1/y=1/20=\ 5\%$	$1{\cdot}05x \ +0{\cdot}05$
„ „	$1/y=1/40=2\tfrac{1}{2}\%$	$1{\cdot}025x+0{\cdot}03$

From these figures it appears that the mixture plotted in fig. 113 as 15 to 1 should, with 5% of air in the generator, be plotted as 15·7 to 1. In the first series of experiments the error is not greater than this, because the charging of

the generator was much less frequent than in later experiments. And, moreover, any discrepant result which was obtained just after charging the generator was repeated. In the second and third series of experiments the volumes of gas used necessitated the more frequent charging of the generator; and consequently the air bears a larger proportion to the gas. Having investigated the cause of the discrepancies, the author feels justified in accepting the pressures exhibited by the smooth curves in the diagrams, especially as the mixtures in the last series have been plotted

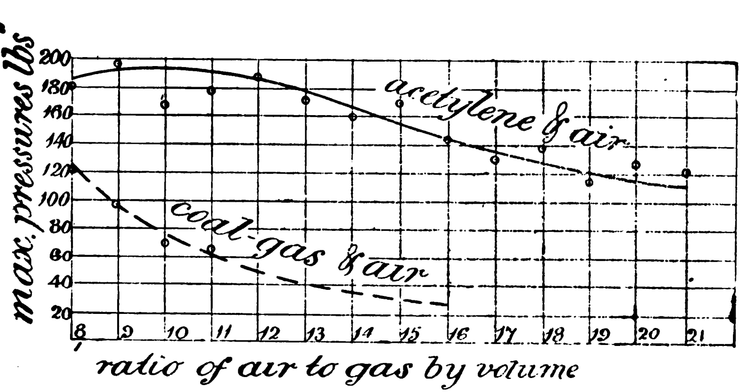


FIG. 114.—Maximum pressures recorded when exploding mixtures of acetylene and air, also coal-gas and air, at two atmospheres initial pressure.

in fig. 115 according to the proportions of air ascertained from the analysis of the products of combustion.

The second series of experiments were made with the mixtures compressed to 15 lbs. per square inch above atmosphere before ignition took place. In each of these experiments the weights of gas used were twice as great as in the first series. The gas required for each combustion was first measured at atmospheric pressure and then driven over into the cylinder. All cocks were then closed, and compressed air was passed into the cylinder, until the pressure gauge

showed 15 lbs. The mixture was then stirred and allowed to stand for ten minutes before ignition was attempted. It was found impossible to keep the pressure at exactly 15 lbs., owing to slight leakages past the valves in connection with the apparatus ; but in all cases the pressure and temperature just before ignition were recorded and allowed for in making the reductions. As the ratio of the gas increased, its leakage preponderated over that of the air, because the undiffused gas in the cylinder occupied the region near the indicator cock as well as that near the inlet cock. The loss of pressure was never more than 2 lbs., and cannot seriously prejudice the results. In series 2 (fig. 114) the maximum

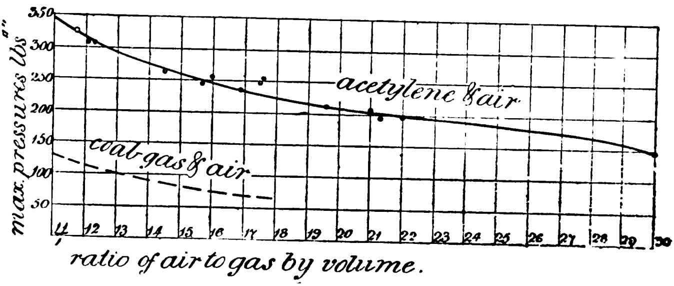


FIG. 115.—Maximum pressures recorded when exploding mixtures of acetylene and air, also coal-gas and air, at three atmospheres initial pressure.

pressure recorded was with a 9 to 1 mixture, and the time of inflammation was 0·02 of a second. The weakest mixture that could be fired was 21 to 1 as measured by the gasholder, and without corrections for air in the generator. Supposing there to be 10 per cent. of air in the gasholder the true mixture would have been 23·1 to 1. It is not likely that there was so much air in the gasholder.

Four mixtures of coal gas in the proportion of 1 and 8 to 11 to 1 were fired at the same initial pressures ; the maxi-

mum pressure of the acetylene explosions was found to be from 1·5 to 2·7 as great as with the corresponding mixtures of coal gas.

In the third series of experiments (fig. 115) the mixtures were fired at 30 lbs. per square inch above atmosphere. The strongest mixture fired was 11·7 to 1, and the weakest 30 to 1. It is probable that higher pressures would have been recorded with stronger mixtures, but it was inadvisable to experiment further in this direction, as the margin of safety of the explosion cylinder was nearing a safe limit. Moreover, it will ultimately be shown that the most economical mixture to use is not in the neighbourhood of 12 to 1, but very much weaker.

In this series the trouble due to leakage was accentuated, and sometimes the pressure dropped 3 to 4 lbs. per square inch during the time the gases were standing for diffusion. Having regard, however, to the fact that in all cases the products of combustion were analysed, and the results plotted according to the presence of air indicated by the analysis, it is probable that a higher degree of accuracy was secured in these experiments than in the former.

352 lbs. per square inch was the highest pressure recorded with an 11·7 to 1 mixture, fired at three atmospheres. The lower limit, namely 30 to 1, gave a pressure of 180 lbs. per square inch. To produce such a pressure with coal-gas a mixture of 9 to 1 was needed.

The time to attain to complete inflammation of the gases was found to be from 13/1000 for strong mixtures to 2/10 of a second for the weakest mixture. It must be noted here that the times recorded by the indicator include a considerable error due to the inertia of the piston and other moving parts of the indicator. And further, that the shorter the time recorded, the greater will be the error due to these causes.

From the diagrams obtained during the explosion of the mixtures the temperatures of the products of combustion have been calculated on the assumption that the laws of

Mariotte and Gay-Lussac are true for these high temperatures and pressures. M. Berthelot, writing on this subject, says, "The laws of Mariotte and Gay-Lussac are hardly applicable in the case of such enormous pressures as those observed in the combustion of powder. With greatly compressed gases the pressure varies with the temperature much more rapidly than would follow from these laws; it approaches the rate observed by physicists in the study of vapours. For a given temperature the pressure is therefore generally higher than that which would be given by calculating according to the ordinary laws of gases. This tends to compensate in the calculation of pressures the contrary influences exercised by the variation in the specific heats."

The theoretical temperature, on the assumption that no heat is transferred to the walls of the cylinder during inflammation of the gases, has been calculated in the following way. The calorific value of acetylene is 1,504 B.T.U's. per cubic foot at 32 deg. Fah. when the water formed by its combustion is not condensed. The weight of the mixtures and the true volumes of air and acetylene are determined from the products of combustion. It remains, then, to find the specific heat of the products. There is some difficulty in this, because of the inaccurate knowledge of the specific heats of gases at high temperatures. Thus the specific heat of steam at 3,600 deg. Fah. (a temperature reached in the experiments) has been given by Mallard and Le Chatelier as 0·68. That of carbonic acid as 0·308, and nitrogen at 0·215. The specific heat of oxygen and carbonic acid are nearly the same at low temperatures, and it has here been further assumed that the specific heat of oxygen at high temperatures is the same as carbonic acid. Even if this assumption be incorrect, the error in the specific heat of the mixtures experimented on is insignificant, because the proportion of oxygen present in the products is very small in comparison to the nitrogen—the chief constituent in determining the specific heat.

The products of combustion of acetylene and oxygen are given by the following equation :—

$$\begin{array}{ccccccc} 2\ C_2H_2 & + & 5\ O & \text{give} & 4\ CO_2 & + & 2\ H_2O \\ (2\ \text{vols.}) & + & (5\ \text{vols.}) & = & (4\ \text{vols.}) & + & (2\ \text{vols.}) \end{array}$$

Thus it is evident that two volumes of acetylene give four volumes of carbonic acid and two volumes of steam. In performing the analysis the steam does not appear because it is all condensed before the gases reach the burette. But the steam exists as steam in the cylinder, and must be included in the calculation for specific heat. It will be noticed that acetylene always produces its own volume of steam and twice its volume of carbonic acid. From the analysis of the products we can easily estimate the volume of steam.

The following analysis was obtained after exploding a 21 – 1 mixture :—

Constituents ...	CO_2	O.	N.	Steam
Volumes (not %)	9·0	8 5	82 5	4 5
Densities	21·9	16	14	8 9
Densities × volumes	197	136	1,150	40
				Total...1,523

Proportion by weight of carbonic acid is—

	197/1,523 and	197/1,523 × 0 308	= 0·040
Ditto oxygen	136/1,523 and	167/1,523 × 0·308	= 0·027
Ditto nitrogen	1,150/1,523 and	1,150/1,523 × 0·215	= 0·163
	36/1,523 and	36/1,523 × 0·68	= 0·016

Total specific heat of one part by weight ... 0 246

From two widely divergent analyses we find in this way that the specific heats work out to 0·246 and 0·249. In working out the results given in the tables the value 0·245 has been adopted; it will be noticed that this value is slightly prejudicial to the efficiency.

The ratio of heat shown on the diagram to that given by these calculations is from 47 per cent to 73 per cent, which

figures may be taken to represent the efficiency of the combustion. With coal-gas the efficiency has never been greater than 50 per cent, and is generally nearer 40 per cent. It is necessary here to bear in mind that the shape of the explosion vessel has considerable influence on the efficiency. The efficiency is greatest when $\frac{\text{surface of containing vessel}}{\text{volume of containing vessel}}$ is a minimum. The efficiency is also very largely influenced by the time which elapses before the inflammation is complete, which is affected by the proportion of diameter to length of containing vessel, the conductivity of the gases, and the walls of their containing cylinder. The conductivity of cast-iron renders the time factor by far the most important in determining the efficiency of the gases experimented upon. Bearing in mind that the time to complete inflammation in the coal-gas mixture is much greater than with acetylene it is not surprising that the efficiency of the latter is much greater.

The phenomena of dissociation have an important bearing on the question of efficiency, as calculated from these experiments. The dissociation of the gases at high temperatures limits the maximum pressure, consequently the higher the temperature the nearer do we arrive to the limit, and the greater is the interference due to dissociation. On the other hand, when the mixture is rich in gas the time for its inflammation is short. This would tend to increase the efficiency, whereas the influence of dissociation tends to diminish the efficiency. Having regard to the order in which the efficiencies work out in Table 3, it would appear that the dissociation of the gases has a preponderating influence, resulting in the gradual reduction of efficiency as the mixtures become stronger in gas. This does not appear to be the case (Table 1) when the mixtures generally contained more acetylene than could be burnt. This influence would, however, not be felt in an internal combustion engine, because the heat due to the re-combination of the gases is evolved during their expansion, which effect is not measured in the above calculations.

It is known that rapid cooling of gases tends to produce permanent dissociation. The analysis of the products of combustion proved that no such permanent dissociation was occasioned by the cooling action of the walls of the cylinder.

Some of the actual explosion diagrams are illustrated in figures 116, 117, 118. With reference to figures 117 and 118, it will be noticed that the rising pressure curves of the stronger mixtures are wavy and irregular. These irregularities have been ascribed by some writers to the effects of dissociation. A great number of diagrams were taken in which no such irregularity appeared, notwithstanding that the temperatures were more than 3,000 degrees F. It is therefore concluded that these effects are due to the indicator. The effect of dissociation is to limit the final or maximum pressure, and to sustain it during the cooling of the gases. It is probable that the effects were produced in the present experiments, but beyond the order of the efficiencies already referred to, there is no evidence of the action of dissociation.

To find the maximum effect produced on a piston by any mixture, the maximum pressure is multiplied by the number representing the volume of air, plus one. Working these values out for each series of experiments, it appears that at one atmosphere a 13 to 1 mixture gives the greatest pressure on the piston. At two atmospheres the most efficient mixture for producing pressure is 15 to 1. At three atmospheres a mixture of 27 to 1. It is probable that with a higher initial pressure weaker mixtures might be found to be even more efficient. In any case it is obvious that the higher the initial pressure the weaker the mixture which gives the maximum effect on the piston, and by increasing the initial pressure to 50 lbs. per square inch it is likely that mixtures of more than 30 volumes of air to one of gas will be found to be the most economical.

It has been thought that great difficulty would be experienced in working a motor on acetylene gas on account

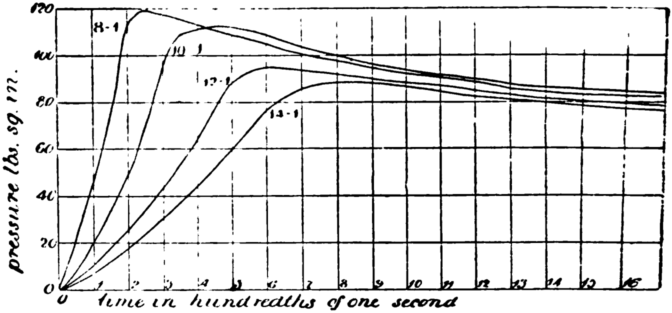


Fig. 116.—Explosion diagrams of acetylene and air ignited at atmospheric pressure.

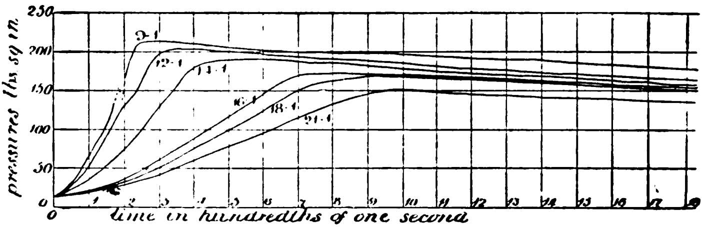


Fig. 117.—Explosion diagrams of acetylene and air ignited at two atmospheres initial pressure.

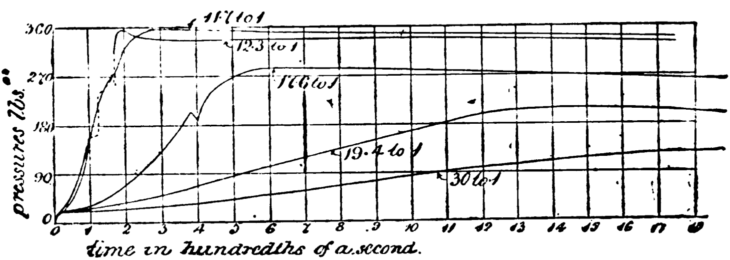


Fig. 118.—Explosion diagrams of acetylene and air ignited at three atmospheres initial pressure.

of the slow diffusion of the gas with the air. The author experienced less trouble with the ignition of acetylene than with coal-gas, when the same time is allowed for the diffusion of each. He therefore anticipates no difficulties in the application of acetylene to motors. It has also been thought that incomplete combustion would lead to a rapid deposit of carbon. After each explosion, when the products of combustion were being discharged from the cylinder, a piece of white paper was placed in front of the small pipe from which the products were escaping at a high velocity. This paper was perfectly clean after all the gases had been discharged from the cylinder. This would certainly not have been the case if a large deposit of carbon took place. There is no doubt that with incomplete combustion occasioned by working with too strong mixtures, there would be a serious deposit. The large excess of oxygen with which it is found possible to explode the gas to advantage is a guarantee that no serious deposit in the cylinder is likely to be met with.

The efficiency of acetylene motors should be higher than that of any heat motor; and the author is of the opinion that this will ultimately reach 35 per cent, chiefly by increasing the speed of revolution. But taking a thermal efficiency of 30 per cent, the consumption of gas will be 6·1 cubic feet per horse power hour. Taking the present price of carbide at £20 per ton, it is evident that the cost of running a motor on acetylene is 2·6 pence per horse power hour. Thus the cost is at present prohibitive of the adoption of this gas in large power stations. But the convenience of the method of generation of the gas renders it of the greatest value for propelling light vehicles. For instance, a vehicle of one ton gross load could run for 10 hours on 60 lbs. of calcium carbide and about 32 lbs. of water for its reduction. About 2½ cubic feet of space would be taken up by the carbide when broken to the gauge of ordinary road metalling (a substance which closely resembles carbide in appearance).

Acetylene for Motive Power.

The characteristics of an explosion of various mixtures of acetylene gas and air are: (1) The great rapidity of flame transmission; (2) the high combustion temperature and (3) the extraordinary energy evolved in the explosion; (4) the low ignition temperature. These points are all in favour of the use of acetylene in combustion motors.

According to the results of experiments made by Mr. H. Schrey on a small motor, we find that 6·35 cubic feet of gas were used per H.P. hour. This confirms the theoretical calculation, based on a thermal efficiency of 30 per cent, namely, that an engine should work with 6·1 cubic feet per hour. With regard to these experiments it was said that twice as much oil was required to lubricate the cylinder as was necessary with coal gas. This may have been due to the unnecessarily large proportion of acetylene gas in the explosive mixture owing to small compression before ignition. The results of the experiments quoted lead to the conclusion that acetylene was not likely to be used to advantage in large installations. We are of the opinion that the experiments were not sufficiently exhaustive to warrant this conclusion, and in view of the explosion experiments elsewhere described, we are strongly of the opinion that with cheap productions of carbide very great use will be made of acetylene for power purposes.

In tests that have appeared from time to time on the use of acetylene in gas engines, the proportions of the gases have been erroneously given. Having regard to the small proportion of acetylene necessary to form an explosive mixture, accurate measurement of the volume is important. But unless the volume of air be measured with the same degree of accuracy, the proportion cannot be determined between the two owing to the expansion of the acetylene in the cylinder and the consequent displacement of the air. The only satisfactory determination of the proportions of air to gas can be made by an analysis of the products of combustion. Any other determination can only be misleading. With the

primitive means at the disposal of the experimenter, in the majority of cases it is probable that he will only be able to ascertain the most economical quantity of gas to be supplied to the engine. This is satisfactory as far as it goes, but it gives no indication of the actual proportion of the mixture, and is no guide for the design of other engines, wherein lies the great value of experimentation.

TABULATED RESULTS.

TABLE NO. 1.

Mixtures of acetylene and air exploded at atmospheric pressure.

Initial Temperature 32 deg. Fah.

Proportion of air to gas ..	18-1	15-1	14-1	13-1	12-1	11-1	10-1	9-1	8-1	7-1	6-1	5-1	4-1
Max. press. lbs. sq. in. ..	54	74	83	83	89	95	103	108	111	112	106	102	101
Efficiency, per cent ..	47	53	56	53	54	58	64	68	71	78	71	70	71

TABLE NO. 2.

Mixtures of acetylene and air exploded at two atmospheres.

Initial Temperature 32 deg. Fah.

Air to gas	21-1	20-1	19-1	18-1	17-1	16-1	15-1	14-1	13-1	12-1	11-1	10-1	9-1	8-1
Max. pres.	121	127	115	138	129	143	171	159	170	168	177	166	196	176

TABLE NO. 3.

Mixtures of acetylene and air exploded at three atmospheres.

Initial Temperature 32 deg. Fah.

Air to gas ..	30-1	22-1	21-1	19·6-1	17·5-1	16·9-1	16·1-1	14·7-1	12·3-1	12·1-1	11·7-1
Max. pres. ..	146	197	207	211	246	236	259	261	308	307	325
Efficiency, per cent ..	48	62	64	61	64	63	62	58	57	57	57

Calorific value of (C_2H_2) acetylene (steam not condensed) = 1504 BTU's. per cubic foot at 32 deg. Fah., and 14·7 lbs. per sq. in.

Ditto (condensed) = 1558 do.

Weight of one cubic foot acetylene at 32 deg. Fah., and 14·7 = 0.0725 lbs.

12·5 volumes of air required per volume of acetylene for its complete combustion.

When acetylene is completely burnt, and the products of combustion analysed, the original mixture of air to gas is given by the following :—

$12{\cdot}5 + 10\,\frac{O.}{CO_2}$ the limiting value of which expression = 12·5 when the O. = 0.

When CO is formed and no oxygen is present in the products, the original mixture of air to gas is given by $\dfrac{6{\cdot}25 + 7{\cdot}5\,\frac{CO}{CO_2}}{\frac{CO}{CO_2} + 0{\cdot}5}$ the limiting value of which = 12·5 when the CO = 0.

$$\frac{\text{Weight of carbide}}{\text{Weight of water required to generate } C_2H_2} = \frac{1}{1{\cdot}77}$$

Analysis of products of combustion of mixtures of acetylene and air exploded at three atmospheres.

Mixtures. $\frac{\text{Air}}{\text{Acetylene.}}$	Constituents by volume per cent.				
	CO_2 per cent.	CO per cent.	O per cent.	N per cent.	Steam H_2O per cent.
11·7-1	13·0	3·2	0·0	79 0	6·5
12·3-1	15·1	0·0	0·0	78·0	8·2
14·5-1	12·4	0·0	2·5	78·9	6·2
16·1	11·8	0 0	4·2	79·0	5·9
17·5-1	10·0	0 0	5·1	79·6	5·0
21·1	8·6	0·0	8·1	78·9	4·3
22·1	9·0	0·0	8·6	79·0	4·5
30·1	6·8	0·0	12·0	78 5	3·4

CHAPTER XXI.

Gas Engine Efficiencies.

We have recorded the historic and modern development of the gas engine, and in so doing we have stated that increased compression before ignition is one of the chief causes in determining the thermal efficiency of the gas engine. We will now examine, from a purely theoretical standpoint, the process of reasoning which leads to this conclusion, and we shall therefrom give other theoretical considerations which will lead the reader to a more complete comprehension of the thermal aspects of the gas engine.

The primary object of our investigation is to ascertain the theoretical thermal efficiency of the Otto cycle gas engine, to which we shall confine our remarks. The thermal efficiency of any heat motor is expressed by the fraction

$$\frac{\text{heat converted into work}}{\text{heat supplied to the engine.}}$$

In testing a gas engine the indicator diagram affords the measure of the heat turned into work, whilst the gas and its calorific value afford a measure of the heat supplied. We shall here treat the subject in a general way, putting H_s for the heat supplied and H_e for the heat rejected from the engine. Using E to denote efficiency we have

$$E = \frac{H_s - H_e}{H_s}.$$

Referring to fig. 119, *ac* is the compression curve of an indicator diagram. It is supposed to be an adiabatic curve, that is, the gas compressed neither loses nor acquires heat. From observation of actual diagrams it will be found that the compression curve approximates very closely to the adiabatic. We shall use the symbols P, V, and τ, to denote respectively absolute pressure, volume and absolute temperature of a mass of gas in the cylinder, and suffixes according

to the point referred to on the diagram. Thus P_a, V_a and τ_a denote pressure, volume and temperature at the point *a* on fig. 119.

The law of adiabatic compression or expansion of a gas is expressed thus $P V^n$ = constant. The value of

$n = \dfrac{\text{specific heat at constant pressure}}{\text{specific heat at constant volume}}$. For the proof of this law we must refer the reader to works on thermodynamics.

The pressure, volume and temperature of the gas are related thus, $\dfrac{P V}{\tau}$ = Constant.

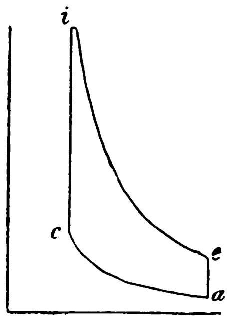


FIG. 119.

Let W be the weight of mixture, C_v its specific heat at a constant volume, the heat produced by the combustion of the gas at the end of the compression stroke will be

$$H_s = W C_v (\tau_i - \tau_c)$$

Similarly the heat rejected at the end of the expansion stroke will be

$$H_e = W C_v (\tau_e - \tau_a);$$

and by substitution $E = \dfrac{W C_v (\tau_i - \tau_c) - W C_v (\tau_e - \tau_a)}{W C_v (\tau_i - \tau_c)}$

$$= 1 - \frac{\tau_e - \tau_a}{\tau_i - \tau_c}$$

This expression may be further simplified by establishing relations between the temperatures denoted, thus :—

$$P_a \ V_a{}^n = P_c \ V_c{}^n$$

$$\therefore \frac{P_a}{P_c} = \left(\frac{V_c}{V_a}\right)^n$$

$$\text{Also } \frac{P_a \ V_a}{\tau_a} = \frac{P_c \ V_c}{\tau_c}$$

$$\text{from which } \frac{P_a}{P_c} = \frac{\tau_a}{\tau_c}\frac{V_c}{V_a}$$

$$\begin{array}{r}\text{Therefore} \\ \text{from above}\end{array} \ \frac{\tau_a}{\tau_c} = \left(\frac{V_c}{V_a}\right)^{n-1}$$

$$\text{Similarly } \frac{\tau_e}{\tau_i} = \left(\frac{V_i}{V_e}\right)^{n-1}$$

But the volumes denoted by V_c and V_i are the same, also V_e and V_a are the same. Consequently the fraction $\left(\frac{V_c}{V_a}\right)^{n-1} = \left(\frac{V_i}{V_e}\right)^{n-1}$, and hence $\frac{\tau_a}{\tau_c} = \frac{\tau_e}{\tau_i}$. To put this in words, we should say, the temperature of the mixture in the cylinder at the commencement of compression is to the temperature at the end of compression, as the temperature at the point of exhaust is to the maximum temperature of the explosion.

This relationship enables us to reduce the expression

$$E = 1 - \frac{(\tau_e - \tau_a)}{\tau_i - \tau_c} \text{ by cross multiplication to}$$

$$E = 1 - \frac{\tau_a}{\tau_c}$$

which may also be written

$$E = 1 - \left(\frac{V_c}{V_a}\right)^{n-1}$$

and

$$E = 1 - \frac{\tau_e}{\tau_i}.$$

Thus we see that the smaller the volume to which the gas is compressed, the less will be the value of the fraction $\left(\frac{V_c}{V_a}\right)^{n-1}$ and consequently the greater will be the efficiency.

CHAPTER XXII.

ENTROPY CHARTS APPLIED TO GAS ENGINES.

ENTROPY diagrams, or as they are usually written $\theta\phi$ diagrams, will be more widely used to represent the changes of heat condition of the motive fluid as they become better understood. The value to the engineer of the abstract quantity termed "entropy" lies in the fact that by plotting changes of the entropy of a gas as abscissæ and corresponding changes of temperature as ordinates, the area under the curve so plotted represents heat units and at once affords a picture of heat distribution which is easy of interpretation. In an indicator diagram by plotting volumes as abscissæ and pressures as ordinates the area under the plotted curve represents work. The thermal efficiency, as we have seen, is

$$\frac{H_s - H_e}{H_s}$$

The $\theta\phi$ diagram enables us to represent these quantities as areas and so present to the eye a picture of the proportion of heat utilised to that supplied to the motor. From this it will be seen that the $\theta\phi$ diagram is of great importance. It affords a complete picture of the heat account of the motor for which the diagram is drawn.

Before attempting to explain the method of working out such diagrams, the mind of the reader unfamiliar with them will be prepared by a careful consideration of the following anǎlagous calculations. Suppose A to lend £500 to B at 5 per cent compound interest payable yearly, and suppose at the same time B to lend A (as a counter loan) £300 at 6½ per cent compound interest also payable yearly, it is required to show by means of a diagram (in which *area* is used to represent money) how much money will be paid by A to B, or by B to A, to balance their accounts between any limits of time within say 10 years from the date of the transaction.

The law of compound interest is

$$M = PR^n$$

where M = the amount (capital + interest).
P = capital.
R = the amount of £1 at the given rate per cent.

From this we can calculate the increments in A's money year by year due to his loan to B of £500 at 5 per cent compound interest.

1	2	3
Year.	What B pays to A in Interest from commencement = Interest to A.	$\frac{\text{Increment to A}}{\text{Time.}}$
1st	£25	25
2nd	£51·25	25·62
3rd	£78·75	26·25
4th	£107·75	26·93
5th	£138·2	27·6
6th	£170·25	28·37
7th	£203·65	29·09
8th	£238 9	29·86
9th	£275·8	30·7
10th	£314·6	31·46

In column 2 the *total* interest from the commencement of the loan is calculated for each successive year up to the tenth year. In column 3 we have divided each increment to A's money by the time (in years) which has elasped when the increment is added. Thus—

$$\frac{25}{1} = 25, \quad \frac{51\cdot25}{2} = 25\cdot62, \quad \frac{78\cdot75}{3} = 26\cdot25, \text{ and so on.}$$

Now if we take the *years* from column 1, and the $\frac{\textit{increments}}{\textit{time}}$ from column 3, plotting the former as ordinates to any

convenient scale of length, and the latter as abscissæ (see Fig. 120), it is evident that the area enclosed between any two horizontal lines and the plotted curve will show the total amount of money which A received from B in the form of interest between the limits of time indicated by the two horizontal lines.

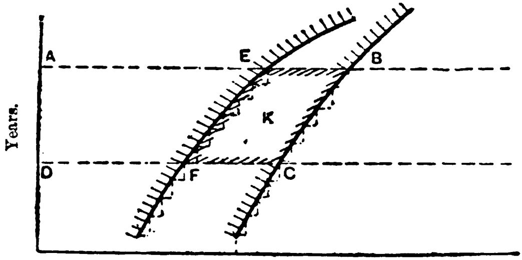


FIG. 120.—Increments divided by time.

Making a similar calculation with regard to B's loan to A we arrive at the following figures :—

1	2	3
Year.	What A pays B in Interest from commencement = Increment to B.	$\frac{\text{Increment to B}}{\text{Time}}$
1st	£19·5	19·5
2nd	£40·2	20·1
3rd	£62·4	20·8
4th	£85·2	21·3
5th	£110	22
6th	£137·7	22·9
7th	£166·2	23·7
8th	£196·5	24·6
9th	£239·2	25·3
10th	£263·1	26·3

This curve is plotted in the same way as the first (see fig. 120). Suppose we wish to find the amount of money transferred from one person to the other between, say, the fifth and the tenth year. Drawing horizontal lines through the points corresponding to five and ten years, we enclose the area C B E F. This, when measured in square inches, and multiplied by the scale of pounds (sterling), shows the balance in favour of A. It will be noticed that the curves converge towards the top. If the financial arrangement existed for a great number of years, a time would come when the "A pays B" curve would cross the "B pays A" curve, and at that time the interest on the two accounts will exactly balance, and for that year A will owe B just as much as B will owe A. After that period A will begin to pay more to B than B to A, and a time will come when the aggregate amount B pays A is equal to the aggregate amount that A has paid B. After that period the whole transaction will be in favour of B, and by means of the diagram the transfer of money may be measured between the limits of any times.

The values in column 3 represent $\frac{\text{increments}}{\text{time}}$ which, to borrow our terms from thermodynamics, are analogous to the "entropy" of a gas, when the increments, instead of being of "money value," are "heat value," and the time is substituted by temperature.

If heat be supplied to a gas contained in a non-conducting vessel of constant volume, the temperature and pressure of the gas will rise simultaneously. The increase in entropy of the gas will be the sum of all the quotients

$$\frac{\text{Very small additions of heat}}{\text{Average temperature at which the addition takes place.}}$$

Expressed in the language of the calculus, we have $d\phi = \frac{dH}{\tau}$ when ϕ represents entropy, H = heat, and τ= temperature.

When heat is supplied to a gas, its pressure, volume, and temperature may vary simultaneously, and the general expression for change of entropy must be deduced from terms of pressure, volume, and temperature. In other words—

Addition of heat may produce	Additional internal energy of the gas, involving rise of temperature and pressure.	+	External effect of work done by the gas expanding between its containing walls.

We know from Joule's experiments how much work is equivalent to a thermal unit, hence we may symbolize the above expression thus—

$$dH = C_v\, d\tau + \frac{p}{J}\, dv \text{ when}$$

C_v = specific heat at constant volume.
τ = temperature.
p = pressure in pounds per square foot.
v = volume in cubic feet.
J = mechanical equivalent of heat.

But $p dv = J(C_p - C_v)d\tau$, and when p is constant $pv = J(C_p - C_v)\tau$.

Hence, substituting for $\frac{p}{J}$ in the above expression, we have

$$dH = C_v d\tau + (C_p - C_v)\tau\frac{dv}{v}$$

$$\text{and } d\phi = \frac{dH}{\tau} = C_v\frac{d\tau}{\tau} + (C_p - C_v)\frac{dv}{v},$$

which is the general expression for entropy of a gas.

When there is no change of volume permitted to take place notwithstanding the addition of heat to the gas,

$$(C_p - C_v)\,\frac{dv}{v} = 0$$

and

$$\frac{dH}{\tau} = C_v\,\frac{d\tau}{\tau}$$

When a change of volume accompanies an addition of heat but there is no change of pressure we have from

$$p\,v = R\tau$$

$$p\,dv = R\,d\tau$$

$$R\,\tau\,\frac{dv}{v} = R\,d\tau$$

$$\frac{dv}{v} = \frac{d\,\tau}{\tau}$$

Substituting $\frac{d\,\tau}{\tau}$ for $\frac{dv}{v}$ in the general expression, we have

$$\frac{dH}{\tau} = Cp\,\frac{d\tau}{\tau}\ .$$

When both the pressure and volume of a gas change simultaneously we have $p\,v^n = p_1\,v_1{}^n$ = constant, where n equals some index depending for its value upon the circumstances. In such a case we must find the particular value of $\frac{dv}{v}$ and insert this in our general expression. To find the particular value of $\frac{dv}{v}$ proceed as follows:—

From $p\,v = R\,\tau$ we have $p = \frac{R\,\tau}{v}$

Substituting the value of p in $p\,v^n$ = constant, we have

$$R\,\tau\,v^{(n-1)} = \text{constant},$$

differentiating we have $d\tau\,.\,v^{(n-1)} + \tau\,(n-1)v^{(n-2)}\,dv = 0.$

$$d\,\tau\,.\,v^{(n-1)} = -\,\tau\,(n-1)\,v^{(n-2)}\,dv,$$

dividing by $v^{(n-1)}$ we have $d\,\tau = -\,\tau\,(n-1)\frac{dv}{v}$

from which $$\frac{dv}{v} = -\,\frac{d\,\tau}{\tau}\,.\,\frac{1}{(n-1)}$$

This particular value of $\frac{dv}{v}$ being now inserted in our general equation we get

$$\frac{d\,H}{\tau} = C\,v\,\frac{d\,\tau}{\tau} - (Cp - Cv)\,\frac{1}{(n-1)}\,\frac{d\,\tau}{\tau}$$

The ratio $\frac{Cp}{Cv} = \gamma$ is of frequent use, and it further simplifies the above expression to write γCv for Cp. Thus we have

$$\frac{dH}{\tau} = Cv\frac{d\tau}{\tau} - (\gamma Cv - Cv)\frac{1}{(n-1)}\frac{d\tau}{\tau}$$

$$= \frac{d\tau}{\tau}\left(Cv - \frac{\gamma Cv - Cv}{n-1}\right)$$

$$= Cv\frac{d\tau}{\tau}\left(1 - \frac{\gamma - 1}{n-1}\right)$$

$$= Cv\left(\frac{n-\gamma}{n-1}\right) \times \frac{d\tau}{\tau}$$

Collecting our results and restating them, we have: When the volume is constant—

$$\frac{dH}{\tau} = Cv\frac{d\tau}{\tau} \text{ and the change of entropy;}$$

$$\phi_1 - \phi_2 = Cv \log_e \frac{\tau_1}{\tau_2}$$

When the pressure is constant—

$$\frac{dH}{\tau} = Cp\frac{d\tau}{\tau} \text{ and the change of entropy}$$

$$\phi_1 - \phi_2 = Cp \log_e \frac{\tau_1}{\tau_2}$$

When the pressure and volume change according to the law pv^n = constant—

$$\frac{dH}{\tau} = Cv\left(\frac{n-\gamma}{n-1}\right)\frac{d\tau}{\tau} \text{ and the change of entropy}$$

$$\phi_1 - \phi_2 = Cv\left(\frac{n-\gamma}{n-1}\right)\log_e \frac{\tau_1}{\tau_2}$$

To apply our knowledge to gas engine testing we require the following data :—

Gases.—Specific heat of mixture before explosion.

Cp and Cv.

Ditto of products of combustion.

Calorific value of the gas.

Weight of gas used per explosion.

An analysis of both the products of combustion, also the constituents of the gas used should be furnished. These are for obtaining Cv and Cp.

Indicator Diagram furnishes temperatures (absolute) and pressures at various points of the stroke.

Also the value of n in $p\,v^n = p_1 v_1{}^n$ should be calculated for both expansion and compression curve.

Engine Dimensions.—Volume of clearance space, and volume swept by piston per stroke.

In order to obviate repetition of well-known calculations we will take the following values as given :—

$Cv = 0{\cdot}1857$, $Cp = 0{\cdot}256$.

$\dfrac{Cp}{Cv} = \gamma = 1{\cdot}379$ for fresh charge of mixture.

$Cv = 0{\cdot}196$ and $Cp = 0{\cdot}268$.

$\dfrac{Cp}{Cv} = \gamma = 1{\cdot}367$ for products of combustion.

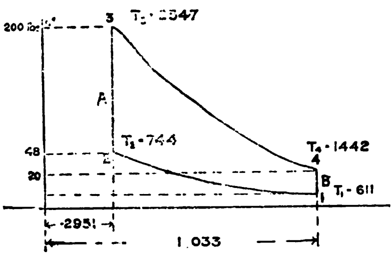


FIG. 121.

The temperature of the gas drawn into the cylinder should be ascertained from the test figures. Indicating the absolute temperature by τ_1, τ_2 &c., as denoted on the diagram (fig. 121), we will find their values from the diagram as follows :—τ_1 is known by the test to be = 611 degrees Fah., and from the relation $\dfrac{pv}{\tau} = \dfrac{p_1 v_1}{\tau_1}$ we can find the values required. These are figured on the diagram. From

$$\frac{\tau_1}{\tau_2} = \left(\frac{v_2}{v_1}\right)^{(n-1)}$$

we can find the value of n for the compression curve, thus

$$\frac{744}{611} = \left(\frac{1 \cdot 033}{0 \cdot 2951}\right)^{n-1}$$

from which $n = 1 \cdot 1573$

Similarly for the expansion curve,

$$\frac{2547}{1442} = \left(\frac{1 \cdot 033}{0 \cdot 2951}\right)^{n-1}$$

from which $n = 1 \cdot 454$

Applying the formulæ deduced, we proceed to calculate the changes of entropy as follows. The suffixes denote the point on the diagram referred to.

$$\phi_2 - \phi_1 = Cv \frac{n - \gamma}{n - 1} \log_e \frac{\tau_2}{\tau_1}$$

Note, Cv is here the specific heat at constant volume of the mixture in the cylinder just before the explosion.

$$\phi_2 - \phi_1 = 0 \cdot 1857 \times \frac{1 \cdot 157 - 1 \cdot 379}{0 \cdot 157} \times \log_e \frac{744}{611}$$

$$= 0 \cdot 1857 \times - \frac{0 \cdot 222}{0 \cdot 157} \times 0 \cdot 1968$$

$$= - 0 \cdot 05168$$

Note the *signs*.

$$\phi_3 - \phi_2 = Cv \log_e \frac{\tau_3}{\tau_2}$$

Note, Cv is here the specific heat at constant volume of the burnt mixture. Although it is not strictly true in practice, the assumption is here made that the combustion is complete when the maximum pressure is reached. In any case a change of specific heat is taking place as the gases burn.

$$\phi_3 - \phi_2 = 0 \cdot 196 \times \log_e \frac{2547}{744}$$

$$= 0 \cdot 2413$$

$$\phi_4 - \phi_3 = Cv \frac{n - \gamma}{n - 1} \log_e \frac{\tau_4}{\tau_3}$$

Cv is here again the specific heat of the products of combustion.

$$\phi_4 - \phi_3 = 0{\cdot}196 \times \frac{1{\cdot}454 - 1{\cdot}367}{0{\cdot}454} \times \log_e \frac{1442}{2547}$$
$$= 0{\cdot}196 \times \frac{0{\cdot}087}{0{\cdot}454} \times -{\cdot}5687$$
$$= -0{\cdot}02136$$

Note the *sign.*

$$\phi_1 - \phi_4 = Cv \log_e \frac{\tau_1}{\tau_4}$$
$$= 0{\cdot}196 \log_e \frac{611}{1442}$$
$$= -0{\cdot}1683$$

If our calculations be correct, the sum of the positive and negative values should be equal, because we have now worked round the diagram.

$+0{\cdot}2413$	$-0{\cdot}05168$
	$-0{\cdot}02136$
	$-0{\cdot}1683$
	$0{\cdot}24134$

Thus our calculations are correct to the fourth figure.

Before plotting the $\theta\phi$ diagram we will re-arrange the values calculated. It will be remembered that our figures relate to *change of entropy*, that is to say that from the point 2 to point 3 of the indicator diagram the entropy is increased by 0·2413. From point 3 to point 4 the entropy is diminished by 0·02136, hence from our zero point (0) on the $\theta\phi$ chart we shall plot the entropy at point 4, as 0·2413 − 0·02136 = 0·21994. Similarly for point 1, 0·21994 − 0·1683 = 0·05164.

From these figures and the corresponding temperatures we have plotted the points 1, 2, 3, 4 of the entropy chart (fig. 122). It will be noticed that the lines joining the points 2, 3, and 4, 1, are curved. In order to get the curve an intermediate value of the entropy change is calculated from the indicator diagram at points A and B. Disregarding for

the moment the other lines on the $\theta\phi$ diagram and confining our attention to the shaded part, we may explain its meaning as follows :—The heat in one pound of the exploded mixture at the time of its maximum temperature is represented by the area 0 2 3 C 0. During the expansion of the mixture in the cylinder there was a loss of heat shown by the area B 4 3 C. This heat would go to raising the temperature of the jacket water.

The area A B 4 1 represents the heat going in the exhaust gases, and the area A 0 2 1 represents the heat passing to the jacket water during the compression of the mixture. The shaded area represents the heat turned into work.

The heat quantities here summarised do not comprise the whole of the heat available, for we find from the test figures that 54·4 units of heat are contained in each charge of explosive mixture entering the cylinder. Further, the weight of the explosive mixture (including the products of combustion left in the clearance space) = 0·0995 lb. Now all our entropy calculations have been worked out for *one pound* of mixture, hence the heat units present in one pound of mixture will be $\frac{54 \cdot 4}{0 \cdot 0995} = 546 \cdot 8$. This heat would produce a rise of temperature of one pound of the mixture of $\frac{546 \cdot 8}{Cv} = 2787$ deg. F. Hence the maximum temperature of the explosion should have been $2787 + 744 = 3531$ deg. F.

Also the temperature at the point of exhaust would be $\frac{\tau_3}{\tau_4} = \left(\frac{V_4}{V_3}\right)^{\gamma - 1}$, from which $\tau_4 = 2365$ deg.F.

Calculating the entropy for the temperatures thus obtained we have

$$\phi_3 - \phi_2 = Cv \log_e \frac{\tau_3}{\tau_2}$$

$$= 0 \cdot 196 \log_e \frac{3531}{744}$$

$$= 0 \cdot 3053$$

From the point 3 to point 4 of the indicator diagram the gas should expand without gain or loss of heat. That is equivalent to saying that the index n in $pv^n = K$ is equal to γ, and under these circumstances

$$\phi_4 - \phi_3 = Cv \frac{n - \gamma}{n - 1} \log_e \frac{\tau_4}{\tau_3} = 0, \text{ because } n - \gamma = 0.$$

There is therefore theoretically only a drop in temperature, but no change of entropy from the point F to E.

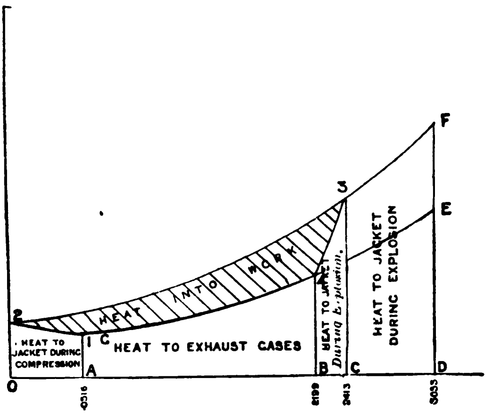


FIG. 122.

From the point E to G we have the change in entropy thus—

$$\phi_1 - \phi_4 = Cv \log_e \frac{611}{2135}$$
$$= -2452$$

From the values worked out for the entropy changes at the higher theoretical temperatures, we plot the curves 23 F, E 41 (fig. 122). The area here enclosed depicts the theoretical possibilities of the engine on the assumption that an unavoidable losses *occur after the compression of the charge*. In an examination of the thermal possibilities of a motor we

ought certainly to commence our calculations from the state in which the gases enter the working cylinder and not after they have been compressed by defective means and under wasteful conditions. It must be carefully noted that the chart (fig. 122) does not depict the theoretical possibilities of the whole cycle, but only that portion of the cycle subsequent to the compression of the charge. In other words, the chart shows that if the combustion and expansion of the charge had taken place without the transmission of heat

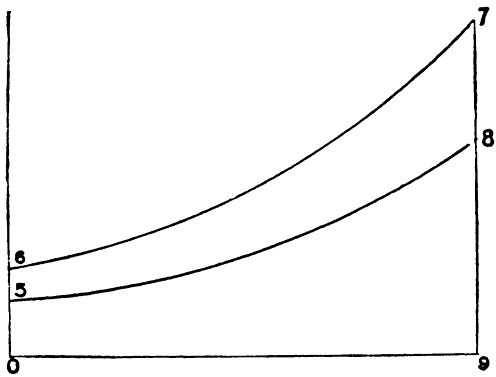


FIG. 123.

through the cylinder walls, it would have been possible to have converted the additional heat area 3 F E 4 into work.

In order to calculate the theoretical possibilities of the whole cycle we will commence with the temperature of the charge on its entry into the cylinder, namely 611 degrees absolute. Calculate the theoretical temperatures at the points 2, 3, 4 of the indicator diagram (fig. 121) from

$$\tau_1 \left(\frac{v_2}{v_1}\right)^{\gamma - 1} = \tau_2$$

The entropy values are on fig. 123. The theoretical maximum

efficiency of the cycle is shown by the proportion which area 5, 6, 7, 8 bears to area 0, 6, 7, 9.

The calculations are rather more tedious when the indicator diagram from which the data are taken has rounded corners, and where the explosion line is not vertical. Let us consider these points in reference to fig. 124.

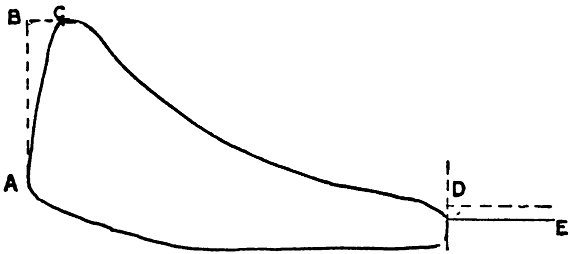


FIG. 124.

Calculate the change of entropy from A to B on the vertical dotted line, then from B to C. With regard to the exhaust, produce the expansion curve and calculate the change at D, then subtract the change of entropy due to the drop in temperature between the imaginary point D and the point E. A sufficient number of points on the curves must be treated in this way to enable the chart to be plotted.

PART II.

PETROLEUM ENGINES.

CHAPTER XXIII.

THE discovery of petroleum in large quantities in Russia and America has materially stimulated inventors to devise means of utilising this enormous latent energy. Before dealing with the oil engines which have attained success, it will be well to discuss the physical properties of the substance known generally as petroleum. The chemistry and the commercial production of petroleum are subjects beyond the scope of this work. Those physical properties, however, which more directly affect the design of oil engines should be carefully studied.

Petroleum has been found at various times in nearly all parts of the world, and has received a variety of names. The most productive American area lies within the boundaries of New York and Pennsylvania. In Canada there has been a large production, and the famous Baku wells of Russia are well known. Petroleum consists chemically of hydrogen and carbon. It is probably of vegetable origin, though this point is vigorously contested by geologists supporting other theories. Crude petroleum is usually of a dark green hue by reflected light, showing red by transmitted light.

An idea of the commercial value of petroleum and the variety of its uses will be gathered from the following table :—

ONE HUNDRED GALLONS OF CRUDE PETROLEUM WILL YIELD UPON FRACTIONAL DISTILLATION :—

		Gallons.	Specific gravity.	Flashing point.
Benzene light oil....	oil engines	1	0·725	– 10 deg. Cen.
Gasolene	oil engines	3	0·775	+ 0 ,, ,,
Kerosene	oil engines	27	0·822	25 ,, ,,
Saliarovi............	lubricating	12	0·870	100 ,, ,,
Veregenni	lubricating	10	0·890	150 ,, ,,
Lubricating oil	lubricating	17	0·905	175 ,, ,,
Cylinder oil	lubricating	5	0·915	200 ,, ,,
Vaseline		1	0·925	....
Residuum............	liquid fuel	14	..	....
Loss........................		10	..	....

The oils used in oil engines are benzene, gasolene, and kerosene, the two former of which are generally known as light oils. They are extremely volatile at ordinary atmospheric temperatures, and are consequently dangerous when carelessly handled. In order that these light oils may be used by owners of oil engines without the inconvenient restrictions imposed by the Petroleum Acts of 1871-9, regulations were issued on November 3rd, 1896, by the Secretary of State. For the purposes of these regulations a light oil is defined as having a *flashing point** of less than 73 deg. Fah. When the flashing point exceeds 73 deg. Fah. the oil is said to be *heavy*, and to this class the above regulations do not apply. Moreover, engines driven by the heavy oils are regarded as safe, and for this reason they will no doubt in time supersede the light oil motors which are at the present time used for driving vehicles. Such oils may be stored in air-tight tanks, all openings from which

* The temperature at which oil commences to give off inflammable vapour, when under ordinary atmospheric pressure, is termed the flashing point.

must be covered with gauze of 400 mesh. No tank may exceed 20 gallons capacity; they must be kept in well-ventilated places. When used in connection with motors for driving vehicles the quantity carried by each vehicle must not exceed 40 gallons.

A rough test of the flashing point may be made in the following way: Take a small quantity of the oil to be tested

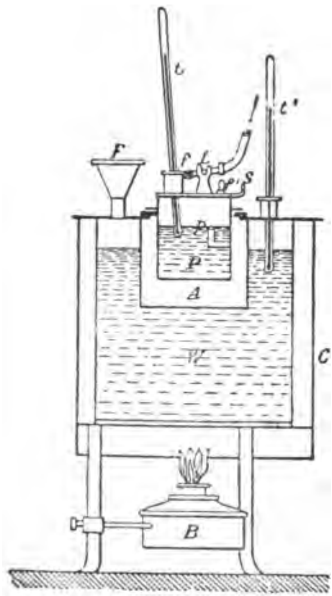


FIG. 125.—Abel tester.

in a metal vessel. Heat the vessel gradually by means of a lighted taper. Apply the light occasionally to the surface of the oil. When the oil ignites immediately put out the flame and take the temperature of the oil with a Fahrenheit thermometer. This will give the flashing point approximately.

In England and Canada an instrument known as the Abel Tester is legally recognised as giving the flashing point. This instrument is shown in fig. 125. The copper vessel W contains water, which is heated by the lamp B to 130 deg. Fah. The oil to be tested is placed in the chamber P, which is surrounded by an air jacket A. A gas flame *f*, outside the chamber P, is arranged to swing round until the flame shoots through a small hole in the cover of the oil chamber, when the hole is uncovered by the moving of the slide S. To test a sample of oil, it should first be cooled to about 60 deg. Fah.; a quantity is then placed in the chamber P, until the pointer *p* is just covered. Light the gas at the nozzle *f*. For each degree rise of the thermometer *t*, the slide *s* is opened, and the flame gently tilted into the oil chamber. A pendulum is usually provided to time this operation to four oscillations, the slide being closed on the fourth swing. When a blue flash is obtained on the application of the test flame, the flashing point of the oil is read from the thermometer *t*.

The specific gravity of shale oils, as sold in Britain, varies 0·78 to 0·85, and the flashing point varies with the density from 76 deg. Fah. to 225 deg. Fah. The following table has been obtained by Professor Robinson, who has carried out an interesting series of experiments, the results of which may be more fully referred to in *The Engineer*, September 11th, 1891.

The chief difficulty in the construction of oil engines is the design of the vaporiser, and in this connection the figures given in the latter columns of the above table are particularly interesting. We see that American Royal Daylight oil boils at a temperature of 144 deg. Cen., but at 215 deg. Cen. only one quarter of the volume of oil treated is vaporised. At a temperature of 230 deg. Cen. 65 per cent by volume still remains. Thus it appears the temperature must be raised far beyond this point in order that the whole volume may be distilled. But it is just here that the designer meets with fresh difficulties, owing to the fact that if the temperature be raised too much decomposition

Properties of Petroleum and Shale Oils used in Britain (Robinson).

Name of oil.	Colour.	Wholesale price delivered in London and Liverpool per gallon.	Specific gravity at 60 deg. Fah.	Specific heat.	Flashing point.		Boiling point, Cen.	Distillation. Volume distilled under 250 deg. Cen.	Distillation. Highest temp.	Distillation. Volume distilled.	Distillation. Time.
		Pence.			Fah.	Cen.	Deg.	Liquid. Per cent.	Liquid. Deg. Cen.	Per cent.	Hrs.
Burning oils.											
American Royal Daylight	Light straw.	4	0·811	0·47	76	24·5	144	25	230	35	3
American ordinary	Light straw.	3¾	0·791	..	75	24	145	29	223	36	3
American Water-white	Colourless.	5	0·780	..	108	42	150	55	216	55	4
American Tea Rose	Light straw.	4½	0·797	..	83	28·3	150	22	243	37	3
Russian ordinary (Russoline)	Light straw.	3½	0·824	0·43	82	27·8	151	30	221	36	3
Russian Lustre	Light straw.	4	0·825	0·45	..	..	..	..	..	..	..
Broxbourne Lighthouse	Light straw.	5½	0·810	0·44	152	66·7	165	First drop	243 270 300	55 90 100	3 2 3
Intermediate oils.											
American mineral sperm	Straw.	..	0·833	..	..	..	195	..	300	5	3
Storrar's Scotch gas oil	Reddish brown.	2½	0·843	..	..	..	195	..	283	5	3
Scotch intermediate shale oil	Clear brown.	2½	0·846	..	..	..	195	..	291	18	2
Light lubricating oil	Clear brown.	2½	0·853	..	225	107	195	..	285	18	2

takes place, and a carbon residue remains, which, if present in large quantities, is quite fatal to the successful action of a vaporiser. The best method of dealing with this class of oils is to raise the temperature to a safe limit, then allow hot air to pass over the oil. In the description of the various methods of vaporising we shall see how this is practically carried into effect. Suffice it to say that in the design of any oil engine the brand of oil to be used must be the first consideration of the designer.

The evolution of an industry may be followed from a careful perusal of the patent office records; but, a history based only upon facts gleaned from this too prolific source of information is likely to be deceptive in that the factors most essential to success are frequently incidentally discovered and escape emphasis. The real steps of progress can only be recognised by an intimate acquaintance with the practical side of the subject and by listening to those who have acted as pioneers.

Many years before the introduction of a practical gas engine some vague patents were granted for the use of spirit as a motive fluid. Its use was probably suggested by the ready distillation of inflammable gas from turpentine and naphtha at low temperatures. But in the early days when coal gas was almost unknown it is clear that the proposal to use spirit was prompted by the desire to obtain a combustible gas. Thus it appears that the oil engine as we now have it was foreshadowed in the earliest attempts to produce a gas engine.

Upon the introduction of coal gas in 1792 a ready form of fuel was placed at the service of inventors, obviating the necessity of vapourising appliances, and thus affording grounds for distinction between the two species of internal combustion motor, now widely known as gas and oil engines. The primary conception of the internal combustion engine was undoubtedly associated with the use of volatile oil. It was, however, necessary that before the more difficult problem of the oil engine could be solved, the simpler gas

engine should become established. It was not until 1876 that we can consider the oil engine to have an existence apart from the gas engine.

Concurrently with the advancement of the gas engine industry, there came into existence a demand for small power light spirit engines for use where no service of gas was attainable. Although these engines were practical in small sizes, the fuel consumption was an item of expense comparing unfavourably with steam plants, and, moreover, the volatile nature of the spirit constituted an element of danger when it was stored in large quantities. It is not surprising that the development of the oil vapour engine has met with legal restrictions on these grounds, nor is it surprising that but little was heard of the oil engine at all until it was made possible to drive it by means of oil attended with less risk.

In November, 1896, the Locomotive on Highways Act came into force and with it there arose the necessity of reconsidering the restrictions hitherto placed upon certain oils. At the time the Act came into force there were many road carriages driven by means of spirit engines, and the regulations issued by the Secretary of State with regard to these were to the effect that mineral spirit which gives off inflammable vapour at a temperature of 73 degrees F. (this being termed the flash point) shall be stored in air-tight tanks of metal, any openings to the atmosphere being covered by wire gauze of not less than 400 meshes to the square inch. We need not mention other restrictions, these being sufficient for our purpose, namely, to indicate a dividing line between what are sometimes spoken of as *light* oil and *heavy* oil. The oil generally preferred for larger power engines has a flash point of about 83 degrees and such oil is for our present purpose termed heavy oil.

Crude petroleum as obtained from the oil wells will upon being slowly heated yield certain gaseous products. The stages of distillation are marked by periods of uniform temperature, therefore, it is noticed that the temperature

rises in steps. Roughly classified the products of distillation at the lower temperatures are benzene, gasolene and kerosene. As the temperature of the residue is raised beyond the point necessary for the evaporation of the kerosene, a series of more dense lubricating oils are driven over into the condensing stills. The specific gravity of benzene and gasolene are, respectively, 0·725 and 0 775, whilst kerosene is of a denser character, being 0 822 specific gravity. The temperatures at which each of these oils give off inflammable vapour is termed the flashing point of the oil. These are as follow: benzene 10 degrees C., gasolene 0 degree C., and kerosene 25 degrees C.

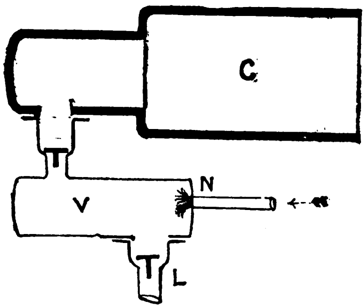


Fig. 126.

It is easy to understand that mere contact with air will bring about the formation of an explosive mixture in the case of the low flash oils, but the difficulties are greatly increased in dealing with the oils of higher flashing point, for if these be heated too much in the endeavour to volatilize them decomposition occurs and a thick and sticky residue will be formed. Owing to this behaviour of the oil there have been many attempts to spray it during the vaporisation and so divide it mechanically into a fine spray, thus assisting the vaporisation by exposing a very large surface to heat. In such cases it was not perhaps fully

realised how readily a drop of oil spreads upon the surface of a heated plate. Many inventors of vaporising appliances have endeavoured to evaporate the oil by heat alone and they have overlooked the important fact that a current of hot air passing over the heated oil will greatly assist the evaporation.

In the successful oil engines of to-day, the vaporisers are designed to utilise the methods of vaporisation above referred to. Figures 126-129 exhibit diagrammatically the four arrangements which are adopted by the various makers

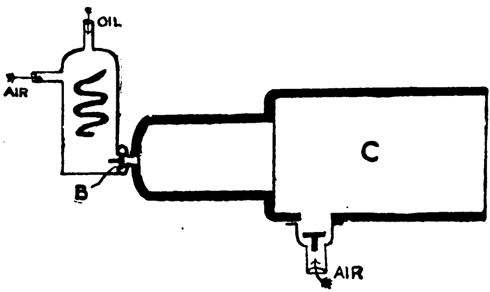


Fig. 127.

of oil engines designed to work with kerosene oil. In fig. 126 C represents the cylinder of the engine, V the vaporising chamber, N the nozzle used for spraying the oil. The chamber V is heated by means of a lamp, and contains an atmosphere of heated air and oil gas. The flow of air through the valve L very materially assists the complete vaporisation of the oil, and a thorough mixing of the charge of explosive mixture takes place before it is drawn into the cylinder. This diagram illustrates in a crude manner the system employed in the Priestman oil engine, which we shall describe more fully later.

In class 2, illustrated by fig. 127, there is no sprayer, the oil being merely allowed to fall upon a corrugated or spiral

surface of heated metal. In order to help the evaporation of the oil, part of the air necessary for forming the explosive mixture is drawn over the surfaces and taking with it the charge of oil gas enters the cylinder. As the piston draws

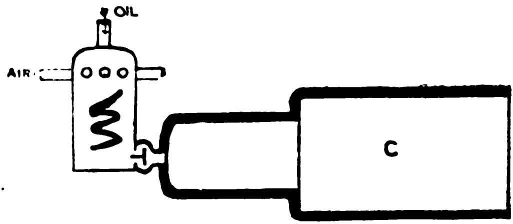


Fig. 128.

in the vapour it also induces the entry of further air into the cylinder through the valve B. The inexplosive mixture mingles with the additional supply of air and on the entire charge being compressed as the piston returns it is exploded

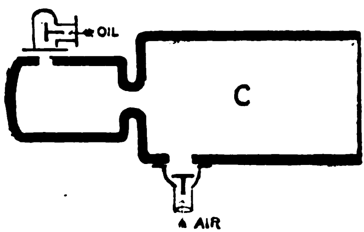


Fig. 129.

and drives the piston forward. Fig. 128 differs from class 2 only in that the whole of the air necessary for the charge is drawn over the vaporising surfaces and enters the cylinder direct. In figs. 126, 127 and 128, there are separate

chambers wherein the oil gas is formed, but in fig. 130 the oil is pumped directly into the space formed at the back of the cylinder, and the vaporising is effected solely by contact with the hot walls of this chamber. The air necessary for combustion is drawn into the cylinder and does not mix with the oil vapour until compression commences.

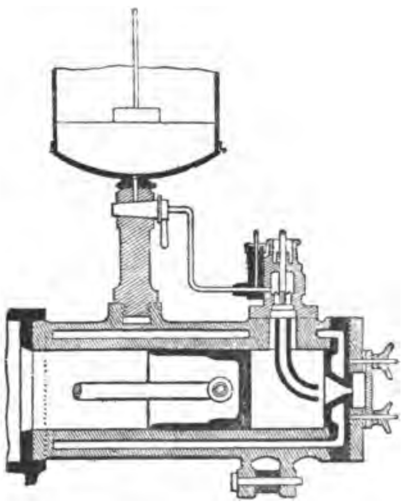


FIG. 130.—Longitudinal section of Spiel engine.

The earliest oil engines worked with the light oils, such as gasolene, which vaporise at ordinary atmospheric temperatures. The method of working was to cause air to pass through scattered particles of the oil, and thus become saturated with oil vapour. It is said that the Lenoir oil engine developed 1 horse power hour by the consumption of 0·9 lb. of oil of 0·65 density.

In 1873 Brayton constructed an engine driven by a safe oil of 0·85 density, the consumption of which is stated to be 2 lb. per horse power hour. In this engine air was mixed with petroleum vapour by forcing the air at considerable pressure through the liquid. The mixture then passed to the cylinder through a network of wire gauze, the latter being kept hot by the exhaust products.

The Spiel engine, a longitudinal section of which is shown at fig. 130, is worked with light oil of specific gravity about

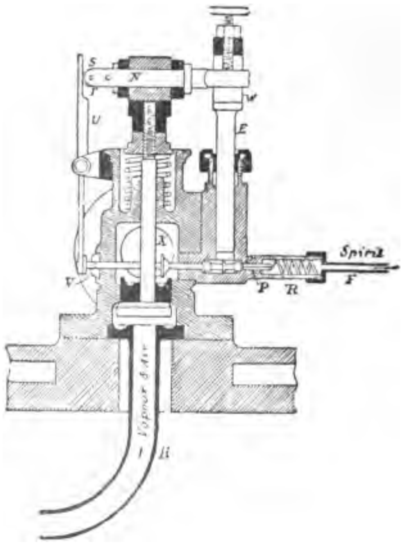


FIG. 131.—Supply valves of Spiel engine.

0·7 The liquid is contained in the upper chamber, from which it runs down the pipe F to the supply valves shown in detail in fig. 131. These valves act in the following way: A spiral spring R forces a double seated valve P towards the end V Spirit, therefore, passes through the open valve and remains beneath the plunger E. A cam shown at M, fig. 132, gives a

vertical motion to the crosshead N, fig. 131. Upon the down stroke of N, the roller S forces the upper end of the lever U outwards, thereby forcing the spindle V inwards, closing

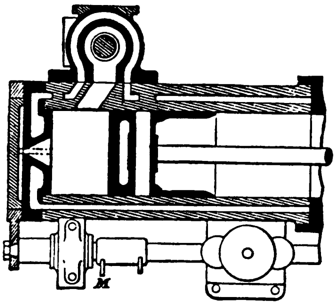


FIG. 132.—Sectional plan of Spiel engine.

the spirit valve P, and compressing the spiral spring R. The down stroke is prolonged until the spirit is discharged by the plunger E against a cone-shaped disc in the chamber X. The

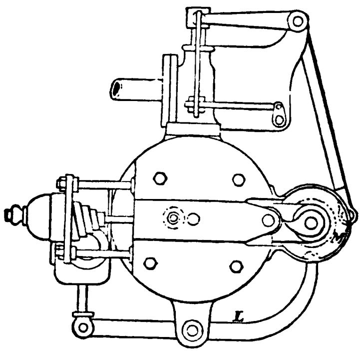


FIG. 133.—End view of Spiel engine.

function of this cone is to spray the spirit and to cause it to mix with air. During the downward movement of the plunger E the admission valve to the cylinder is opened, and

the piston, on its outward stroke draws the spirit vapour and air down the pipe H. Ignition is caused by a moving slide, shown in figs. 130, 132, 133. The action of the slide is very similar to that of the early Otto slide ignition engines.

Simplex Carburator.—Messrs. Delamare-Devoutteville and Malandin have adopted an arrangement for supplying gas

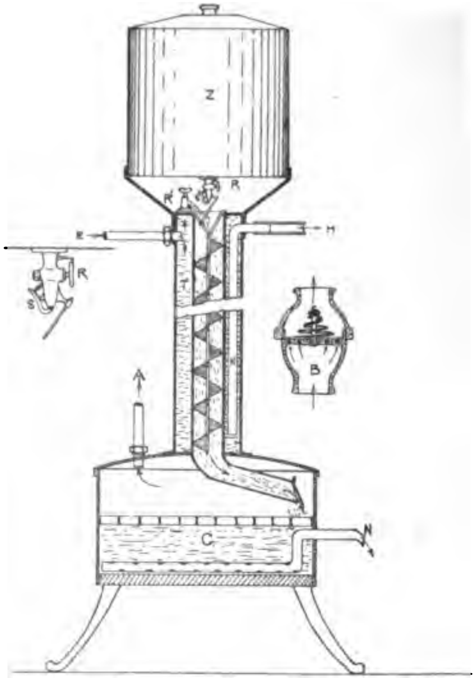


FIG. 134.—Simplex Carburator.

to their Simplex gas motors where a supply is otherwise unavailable. Fig. 134 shows a sectional elevation of this carburator. The spirit (density about 0·7) is contained in the tank Z. The pipe E conducts hot water from the engine

jacket to a cylindrical casing T, whence it passes away through the pipe H. The central chamber contains a spiral wire brush, down which the spirit drops. Through the cock R a quantity of hot water is allowed to pass down the spiral; this assists in heating the spirit to completely vaporise it.

The water falls into the chamber C, whence it escapes through the pipe N. A perforated cork floats on the water, thus preventing the suction of water spray into the cylinder through the suction pipe A. The valve B is attached to the suction pipe A, and is used to prevent the return flow of high-pressure gases from the cylinder to the carburator. The supply of spirit is controlled by the action of the governor on the valve S.

The use of petroleum for firing boilers containing water has been successfully carried out, and economy has also been sought by evaporating spirit instead of water in the boiler. This arrangement is attended with considerable danger unless very carefully handled; at the same time it is true that certain thermal advantages result from the evaporation of spirit instead of water, on account of the smaller quantity of heat supplied to the spirit being rendered latent. These arrangements cannot be regarded as internal-combustion engines, and will not therefore be described.

CHAPTER XXIV.

It is not our intention to traverse the early history of the oil engine. We shall at once, therefore, enter upon a description of the engines which have been made within the last few years. It will be understood that the main difference between the gas and oil engine is the addition to the former of some means of causing the oil to form an explosive mixture with air. This being so, our attention need only be directed to this matter; indeed, many makers are now building gas engines which may with very little trouble be converted into oil engines. This necessitates the addition of a vaporising chamber, which need be no encumbrance when the engine is worked by a supply of gas.

It has been the object of many inventors to construct an engine capable of working with heavy petroleum oils, instead of petroleum spirit. The inflammability of spirit, even when a light is applied at a distance of 2 in. or 3 in. from the spirit, renders its use highly dangerous; whereas petroleum oil will extinguish a lighted taper when, at ordinary atmospheric temperatures, the taper is plunged into the oil.

Although many attempts were made to utilise the heavy oils, none led to practical results until Messrs. Priestman, of Hull, having acquired Etève's patents, persevered with the constructive details, and were thus able to place a satisfactory engine upon the market in the year 1888.

The author obtained much interesting information, and was permitted to take photographs of engine details, some of which are referred to later. The story of Messrs. Priestman Brothers' pioneer work is here told as we received it.

Priestman Brothers' first connection with internal combustion engines was in the year 1885, when they took up the manufacture of the "Eteve Spirit Engine."

MM. Eteve and Lallement, of France, had shown an engine working in London with oil *spirit*, and Mr. J. J. R.

Humes, of London, had made a series of tests clearly demonstrating that motive power was to be obtained from a hydrocarbon engine.

In America, Brayton had spent a fortune in the endeavour to perfect such an engine, without success; other attempts had failed, and as there was no such engine on the market, a great future was predicted for a reliable petroleum engine, if this could be produced.

At this time it seemed a matter of comparatively small importance whether power was obtained from the volatile products of petroleum or from the ordinary petroleum of commerce. Many months of incessant work were confined to the perfection of the petroleum spirit engine, which was nearly accomplished, when it was thought that motive power obtained from the use of benzoline and similar highly explosive products could never be considered safe. Very limited quantities only could be stored without a license, the premium for fire insurance was high, and it became evident that if *oil* could be used instead of spirit, a very much wider field for the use of such an engine would be opened out.

This being so, the opinion of one of the leading chemists of the day was obtained, who had given special attention to hydrocarbons, but he discouraged a proposal to make any attempt in this direction, as, from his experience, it was sure to fail. Attention was therefore, for a time, again confined to further perfecting the spirit engine, and later an attempt was made with common oil, when a few turns of the crank were obtained.

The expert in oils already referred to was informed of this, and further advice was asked for, when he stated that he could not advise, as, so far as was known, power had never been obtained from oil, and if this were done, the method by which it could be accomplished had yet to be found out.

Many months of hard work followed, attention being directed first to one detail and then to another, resulting

only in fitful and erratic behaviour of the engine, followed by long periods when the engine would scarcely turn one revolution.

FIG. 135.

It was in the spring of 1887, after ceaseless attention had been given to obtain an atomizer, or spraymaker, capable of mixing air and oil so perfectly as to avoid the deposit of carbon in the working parts, that some success was obtained.

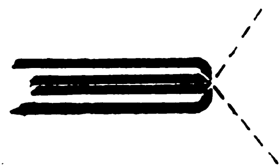


FIG. 135A.

The accompanying illustrations, figs. 135, 135A, 135B, and 136, made from the author's sketches and photographs, may be taken as representative of the evolution of the nozzles, which were tested in every conceivable

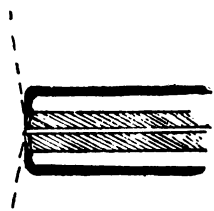


FIG. 135B.

manner, when, at last, it was proved that a sprayer fitted with an inverted nozzle, so designed as to throw air at a considerable pressure upon a jet of oil in an opposite direction discharged at a similar pressure, gave

results hitherto unattained. The spraymaker so fitted proved very superior to anything previously tried, and it was felt that real progress had at last been made. Experience confirmed these impressions, and the maturing of other details followed.

After some lengthened runs had been made, it was decided to ask Lord Kelvin, then Sir William Thomson, if he would test the engine and report on its working. Of such interest

FIG. 136.—Illustration of the Evolution of the Inverted Spraying Nozzle.

was the engine that Lord Kelvin tested and reported upon the running of several Priestman oil engines, from which the following is taken :—

"I have inspected Priestmans' petroleum engines at their works, where I found six engines all working with common petroleum, of gravity about 800.

"I made careful tests on a 6 h.p. engine. After seeing it started and stopped several times, and kept running on the brake for an hour at 7½ h.p., and for two hours at 6 h.p., without measuring the oil, I gave it exactly an hour's run with the brake load slightly more than for 6 h.p., and with

arrangements to measure the oil accurately. The quantity of oil used was at the rate of (see footnote) 1·69 lb. per hour per brake horse power, which seems to me remarkably good economy, considering the great difficulties which had to be overcome in using the combustion of oil directly as a motor. It must be noted that these results refer to the horse power of work actually done externally by the engine, and not merely to 'indicated horse power,' which in the steam engine, and still more in the gas engine, falls short of true horse power by a large difference.

"Messrs. Priestman's engines are simple in construction, and there are few working parts liable to get out of order.

"By a new and effective mode of regulating the supply of vapour to the cylinder, combustion so perfect is obtained that deposit of carbon in the cylinder and passages is most satisfactorily obviated, as I have myself verified by careful examination.

"As the engine is governed by reducing the charge admitted into the cylinder, instead of cutting off the supply, the explosion takes place with great regularity, thus securing steady running with or without load, and with varied loads, which judging from my own experience of the irregular running of gas engines, running at anything less than full load, is a very important advantage.

"The piston requires no oiling, as the vapour admitted into the cylinder lubricates it sufficiently. As the engine has all the advantages of a gas engine, without being dependent on gas works and a gas supply, it is available for many important applications, from which the gas engine is precluded."

At Nottingham, a few months later, the Royal Agricultural Society, upon the recommendation of the late Sir Wm. Anderson, awarded the Priestman Oil Engine their medal, being the first ever given for an internal hydrocarbon engine, and the following year a second medal was awarded for an engine of a different type. In 1890 the Society's Special Prize was awarded to the Priestman Oil

* This consumption has been greatly reduced since Lord Kelvin made these tests.

Engine at the competitive trials held at Plymouth, when the judges, Professor Unwin and Mr. Dan Pidgeon, C.E., reported as follows :—

"It may be said at once that the working of the Priestman engine was altogether satisfactory, and it worked with an economy of fuel almost unprecedented. So far as could be judged from a trial extending altogether over nearly six hours, the action of the engine was faultless.

"The Priestman engine works with about as much fuel as the very best large steam engine, with about one-eighth as much fuel as a small non-condensing engine, and with an economy about as great as that of a gas engine with Dowson gas producer.

"Taking, however, the more scientific comparison of the heat value of the fuel, the Priestman engine is better than the best large condensing steam engines, six times better than Messrs. Turner's steam engine, and very slightly worse than an Otto engine using Dowson gas."

Since 1890 experience has suggested improvements of considerable importance.

All the larger engines are now fitted with a self-contained starter, which is a very reliable mode of putting the engine in motion, without turning round the wheel.

Ignition by tube, when preferred to that of the electric igniter, is supplied, and as a further means of keeping the internal parts of the cylinder and valve chest clean, water is injected into the combustion chamber, with the result that engines will run for very long periods without it being necessary to clean the cylinder, piston, valves, &c.

In the Priestman engine the idea of spraying oil mingled with air into a heated chamber, called the vaporiser, was first brought forward.

Although it is now well known that oil may be vaporised without being finely divided into a spray and mixed with air, it is a noteworthy fact that the development of this class of motor proceeded on the lines indicated by Messrs. Priestman, and very great credit is due to this firm for their

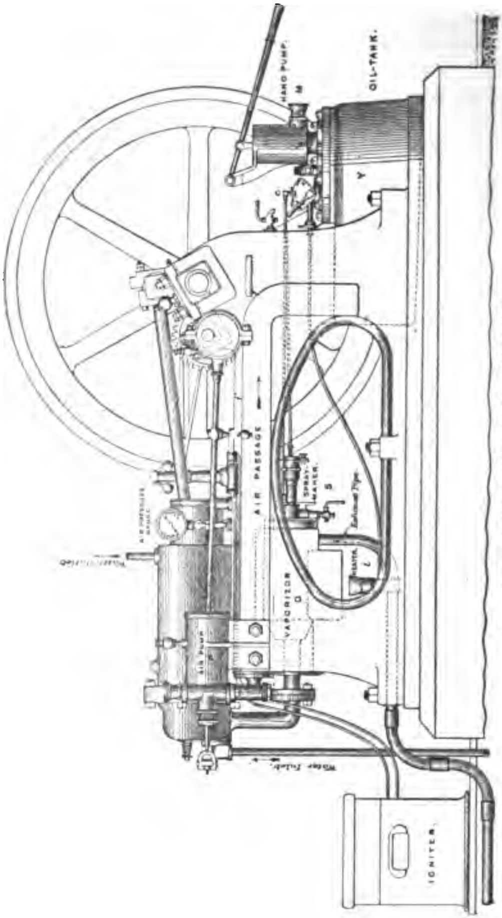


FIG. 137.—PRIESTMAN OIL ENGINE.

persevering efforts, which have resulted in the production of a motor of great economy and usefulness. The following description, together with the illustrations, will suffice to explain the working of the Priestman engine.

The oil is contained in a cast-iron tank Y, fig. 137, formed in the bed of the engine. The oil, with a supply of air under a pressure of about 8 lb. per square inch, is carried to the spray maker S; here they mix and break up into a fine vapour. This is then passed through a heating chamber, called the vaporiser (which latter is warmed by a lamp *l*

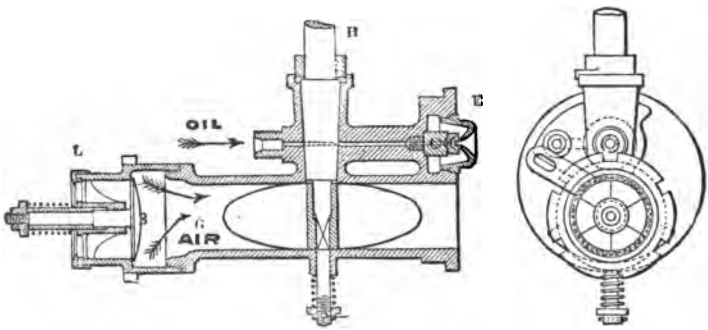


FIG. 138.—Section and End View of Spray Maker of Priestman Engine.

before starting, and is kept hot, when running, with the exhaust products). From this chamber the vapour is drawn into the engine cylinder, compressed by the piston, and ignited by an electric spark, or lamp and hot tube, as preferred. With the Edison L'alande system of creating the spark, an engine can be run for 500 to 1,000 hours without any attention being given to the two small cells. The governing is effected by regulating the quantity of oil, instead of by entirely stopping the supply. The piston is lubricated by the deposit of oil on the surface of the cylinder.

Fig. 138 shows a section of the spraying nozzle, and the wing valve for controlling the air supply. Oil is forced into

the nozzle E by means of compressed air in the oil tank. On its way to the nozzle the oil passes through a tapered hole in the plug H, to the lower end of which the wing valve G is attached. The plug H is controlled by the governor, and it will be obvious that both the oil and air are simultaneously throttled, thus keeping the mixture uniform The nozzle E, a larger view of which is shown in fig. 139, is of special construction, and is the result of a long series of experiments. Oil enters the nozzle through the central opening, and the air is forced in through the annular space. It is an important feature in the construction of the nozzle that the air current is directed against the flow of oil. In the earlier

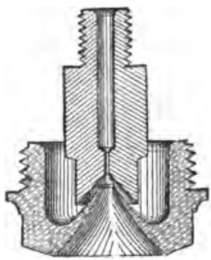


FIG. 139.—Spraying Nozzle, Priestman Oil Engine.

nozzles used by Messrs. Priestman the air and oil were made to flow in the same direction, where they mingled together. The form was, however, gradually altered, until it was found to give the best spray when made as in fig. 139.

For the ignition of the charge a simple bichromate cell may be used, with one induction coil.

The charge recommended for the battery is—

8 parts by weight of bichromate of potash.
6 parts by weight of sulphuric acid.
80 parts by weight of water.

This charge will, if made up in ounces, last for about 30 hours, at a cost of about 6d.

The following figures have been obtained by Professor Unwin in a systematic test of a Priestman oil engine:

Loads.	I.H.P.	B.H.P.	Mechanical efficiency.	Oil per B.H.P. hour. Pounds.	Brand of oil used.	Mean effective pressure. Pounds sq. in.	Revolutions per minute.	Temperature of vaporising chamber.
Full load..	7·40	6·76	0·91	0·946	Russolene	41·38	207·73	264·5
Half load..	4·70	3·62	0·76	1·381	Russolene	25·48	214·29	254·0
No load...	0·889	..	..	I.H.P. 5·734	Russolene	5·51	187·3	260·5
Full load..	9·36	7·72	0·82	0 842	Royal Daylight	53·2	204·33	268·0

The dimensions of the engine from which these figures were obtained are as follow:

Diameter of cylinder	8·51 in.
Stroke	12·0 in.
Clearance in percentage of working volume..	53 per cent.
Volume of air-compressing pump per stroke	0·0329 cu. ft.
Diameter..	5·12 in.
Diameter of flywheel	4·61 ft.
Weight of engine without flywheel.........	26 cwt.
Weight of flywheel	10 cwt.

The oils used in the above trials yielded the following constituents:

	Russolene. Per cent.		Royal Daylight Per cent.
Carbon	85·88	....	84·62
Hydrogen	14·07	....	14·86
Oxygen, &c., by difference......	0·05	....	0·52
	100·00		100·00

Density at 60 deg. Fah......	0 822	0·793
Flashing point (Abel test)...	86 deg. Fah.	77 deg. Fah.
The calculated total calorific value	21,180	21,490
The calculated calorific value when steam formed is not condensed	19,957	20,198

The following is a résumé of the results of a trial conducted by Professor Unwin and Mr. D. Pidgeon, at the Royal Agricultural Show, Plymouth, in 1890 :—

PRIESTMAN OIL-ENGINE TRIALS.

Loads.	I.H.P.	B.H.P.	Mechanical Efficiency.	Oil per B.H.P. hour. Pounds.	Brand of oil used.	Mean effective pressure. Pounds sq. in.	Revolutions per minute.
Full load	5·24	4·49	0·85	1·066	Broxburn oil	83·96	179·5
Half load	3·21	2·36	0 73	1·460	Broxburn oil	20·65	180·3

ANALYSIS OF BROXBURN OIL.

Carbon	86·01	per cent.
Hydrogen	13 90	„
Undetermined	0 09	„

Calculated calorific value of Broxburn oil, when steam formed is not condensed, 19,700 British thermal units.

The Priestman oil engine has been successfully used for pumping, electric lighting, fog signalling, and for driving launches and barges. For the latter purpose Messrs. Priestman supply a patent reversible screw propeller, the blades of which can be inclined to drive ahead or astern without stopping the motor.

The Samuelson Oil Engine (*Griffin's Patent*).—This engine made by Messrs. Samuelson and Co. Limited, Banbury, and

the Priestman engine, are the only two manufactured which work with the spray maker. The oil sprayer is shown in fig. 140. Its action is somewhat different from the Priestman nozzle, inasmuch as the oil is sucked up into the annular space surrounding the air nozzle by the action of the horizontal air jet; whereas in the Priestman engine the

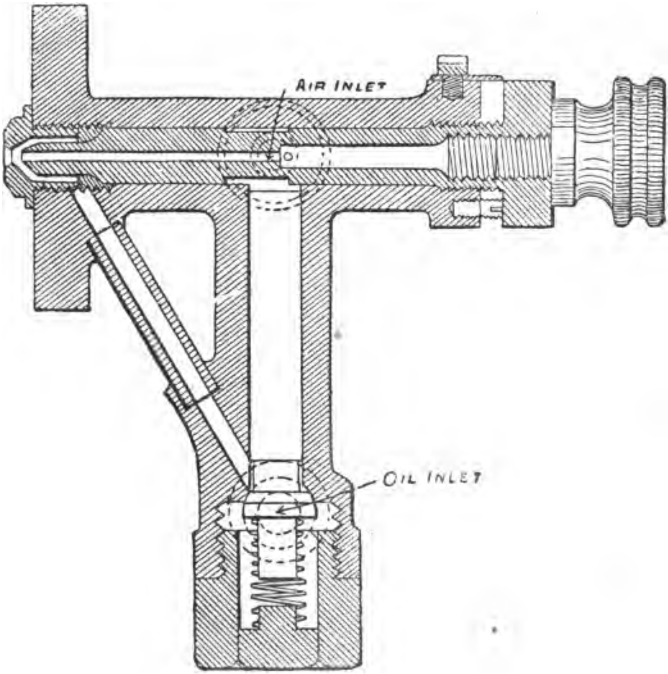


FIG. 140.—Griffin Patent Oil Sprayer.

oil occupies the central nozzle, and the air meets it at the orifice. The air pressure is maintained by an eccentric-driven pump, at about 12 lb. above atmosphere. The vaporiser is a long corrugated chamber, heated by the

exhaust products. The governor is of the ordinary centrifugal pattern, and acts by cutting off the air supply, at the same time preventing the opening of the exhaust and admission valves. This method of governing has the advantage that the cylinder is not cooled by the admission of cold air when the engine is missing fire.

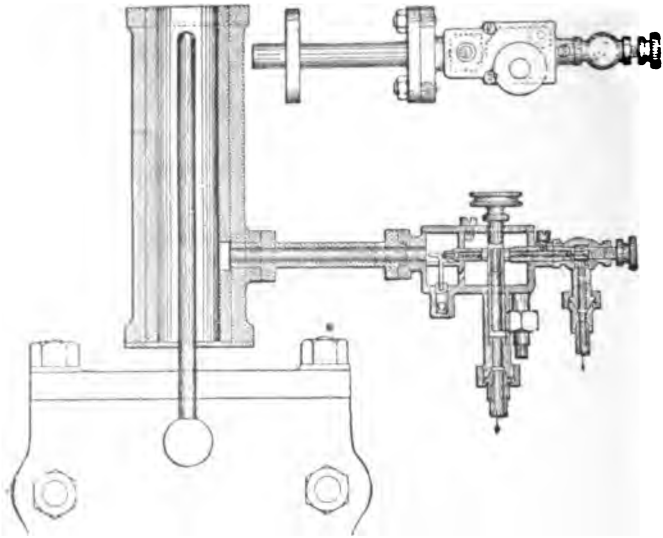


FIG. 141.—Samuelson's Igniting Arrangement.

The igniting arrangements are shown in fig. 141. An air jet plays upon a piece of bent wire, which latter dips into a chamber filled with oil to a constant level. The air jet draws a film of oil up the wire, carries the oil forward in the form of spray, and the mixture burns around the ignition tube. In starting this engine a hand pump is used to compress air for the ignition spray and for the vaporiser sprayer. Both are lighted, and the engine allowed to stand for about ten minutes, until the vaporiser is hot enough to

work. The length of the ignition tube has apparently no little influence upon the form of card obtained, due no doubt to the fact that the charge does not get compressed into the hottest part of the tube when the latter is too short.

The Hornsby-Akroyd Oil Engine, illustrated in figs. 142, 143, 144, manufactured by Messrs. R. Hornsby and Sons Ld., Grantham, England, is constructed to work upon the Beau de Roches cycle, being very similar in appearance to a gas engine.

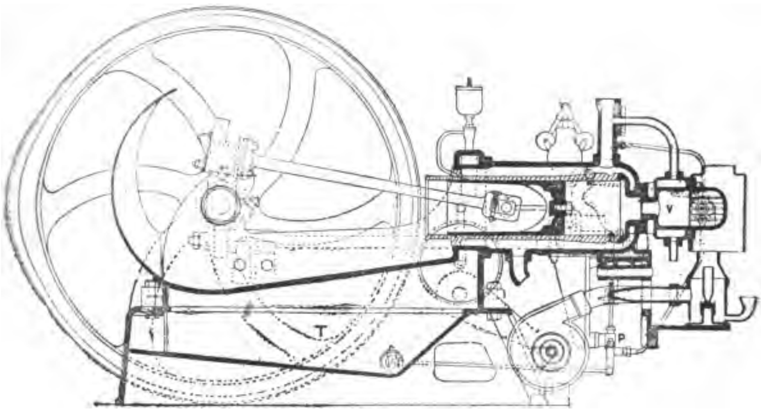


FIG. 142.

The vaporiser is a special feature of this engine and distinguishes it from all other types by not requiring either a hot tube or electric spark.

The vaporiser V (see fig. 142) is a bottle-shaped chamber or extension of the cylinder, being connected with the cylinder only through a neck or contracted passage.

It is partially water-jacketed in the medium and large size engines, and is heated by a lamp and fan-blower when first starting the engine; afterwards it maintains itself at a temperature high enough to cause ignition of the oil vapour and air.

The oil is pumped from a tank T, formed in the base of the engine, by a small plunger pump P into the hot vaporiser during the air suction or charging stroke. The oil is then vaporised by the hot walls of the vaporiser and mixes with the products of the combustion remaining from the previous explosion.

The air is not drawn into the vaporiser but directly into the cylinder, and on the compression stroke is forced into the vaporiser through the neck and there mixes with the vapour contained in it.

At first the mixture does not contain sufficient oxygen for combustion, but at the end of the compression stroke the

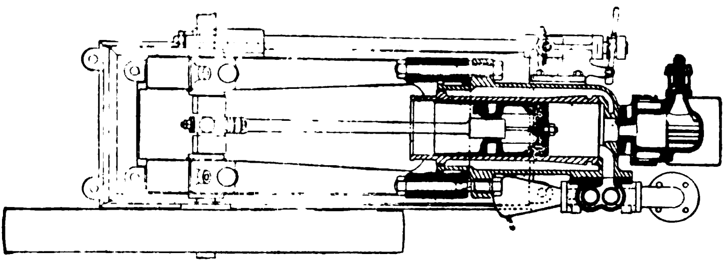


FIG. 143.

mixture attains the proper explosion proportions and is ignited by the hot walls of the vaporiser.

Another important feature of this engine is that it is claimed to work satisfactorily with heavier oils than other engines; even with some oils weighing up to 9½ lbs. per gallon if the engine be suitably adjusted for the purpose; but the oils recommended as being the most powerful and giving the most economical results are refined Russian or American petroleum oils, having a specific gravity of from ·79 to ·825, and a flashing point (Abel's close test) of from 74 deg. to 83 deg. Fah., or following these any of the well-known brands of refined Scotch oils having a specific gravity of about 8·1 and a flash point of 225 degs. Fah. Abel's close test;

also the heavy crude petroleum oil or Astatki, if properly adjusted.

The oil pump previously mentioned is connected to and actuated by an air-lever and forces the oil immediately prior to its entering the vaporiser through a valve box attached to the vaporiser in which box are two spring valves, one horizontal, the other vertical. The oil enters the valve box

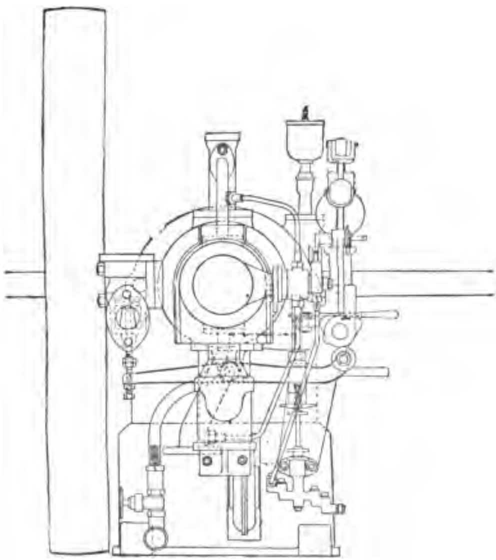


FIG. 144.

by way of the horizontal valve which is opened by the pressure of the pump and then flows through the spraying jet into the vaporiser.

The regulation of the engine is effected by a Porter governor opening the vertical or overflow valve when the

speed is too high and causing the oil pump to return the oil to the tank.

A regulation handle is also provided by means of which the vertical valve may be opened or shut and the supply of oil intercepted, this being the method of stopping the engine. If it is known that a light load only is to be dealt with, more steady running is secured by altering the stroke of the oil pump, for which due provision is made.

The air and exhaust valves are worked in the same manner as in a gas engine, *i.e.*, they are opened by separate levers, each actuated by its own cam, mounted on a horizontal shaft which is driven from the cam shaft by skew gearing, so geared as to make one revolution to two of the crank-shaft.

The cylinder is water-jacketed as in the gas engines.

The general arrangement of the working parts of the Portable engine made by Messrs. Hornsby is precisely the same as that of the fixed engines and will need no further description. It is mounted on a wrought-iron frame, running on four travelling wheels. The oil and water tanks are beneath the frame work, and a circulating pump is used to circulate the water from this tank through the jacket, and also through vertical boards in a cooler at the front end of the frame.

The exhaust gases are carried into a silencer formed on the top of the cooler, from thence through a blast pipe into the chimney, and induces a current of air through the boards in the contrary direction to that of the water.

The Hornsby engine was awarded the first prize in competition with nine other makers at the Royal Agricultural Show at Cambridge. According to the report of the trials, written by Professor Capper, this engine ran without hitch of any kind from start to finish. One attendant only was employed all through the trials, and he started the engine easily and with certainty, after working the hand blast to the lamp for eight minutes. The longest time taken to start was nine minutes, and the shortest seven. The revolutions

were very constant, and the power developed did not vary one-quarter of a brake horse power from day to day. The oil consumption, reckoned on the average of the three days run, was 0·919 lb. per brake horse power hour, including the oil used for the starting lamp. The oil used was Russolene, sold at Cambridge at that time for 3¾d. per gallon. The cost of oil per brake horse power hour is therefore about ¾d. At half-load the oil consumption was found to be 1·49 lb per brake horse power hour.

Trials carried out on January 4th, 1899, By W. Robinson.

General Results.

Trials of "Hornsby-Akroyd" 25 B.H.P. Oil Engine.

Power.	Normal.	Two-thirds.	Light.
Duration of trial (hours)	8	8	1
Mean revolutions per minute	202·56	202·4	201·5
Mean explosions per minute	101·3	101·2	100·7
Mean effective pressure, lbs. per sq. in.	43·4	31·2	6
Mean brake H.P.	26·74	17·96	0
Mean indicated H.P.	31·03	22·38	4·28
Engine friction H.P.	4·29	4·42	4 28
Mechanical efficiency of engine, per cent	86	80	..
Oil used per hour (lbs.)	19·75	16·75	5·75
Oil used per I.H.P. per hour (lbs.)	0·63	0·74	1·34
Oil used per B.H.P. per hour (lbs.)	0·74	0·91	..
Inlet temperature of cooling water, Fah.	91°	101°	110°
Outlet temperature of cooling water, Fah.	138°	130°	142°
Loss per minute by jacket B.T.U.	3172	..	..
Cost of oil used (pence) per effective or brake H.P. (hour)	·359d.	·44d.	..

BRAND OF OIL USED "H. V. O."

A refined Russian petroleum called Russoline was used by the engine in all the trials. Samples were taken and examined with the following results :—

Specific gravity at 60 deg. Fah.......... ·825
Flash point (Abel close test) .... 90° Fah.
Composition by Analysis { Carbon ... 86·5 per cent. / Hydrogen 13·5 „

FRACTIONAL DISTILLATION.

"H.V.O." Russoline began to distil at 115° Centigrado.
Spirit driven off below 150 deg. C......................15 per cent.
Normal kerosene distilled 150 deg. to 270 deg. C. 15 per cent.
Residue ....10 per cent.

DETERMINATION BY EXPLOSION IN BOMB CALORIMETER HEATING VALUE OF FRACTIONS.

		B.T.U. per lb.
Spirit below 150 deg. C.	15% × 19840	= 2976
Normal kerosene 150 to 270° C.	75% × 20385	= 15289
Residue	10% × 19969	= 1997
Total heating value of the fractions		= 20262 B.T.U.*
And the original oil		= 20286 B.T.U.*

Total calorimetric value found by Bomb calorimeter.................................... 20286 B.T.U. per lb.
Effective heating value (after deducting latent heat of steam)..................... 19100 B.T.U. per lb.

DIMENSIONS OF ENGINE.

25 B.H.P. Hornsby-Akroyd Oil Engine.
The following are the dimensions of the engine measured at the trials :—

Diameter of cylinder............ 14·5 inches.
Stroke of piston .... 17·0 inches.
Area of piston.....................165·13 sq. inches.

(1) Effective circumference of one brake wheel...18·995 feet.
(2) Effective circumference of other brake wheel...18·966 feet.
(1) Wheel constant per revolution per lb............ 0·0005756.
(2) Wheel constant per revolution per lb............ 0·0005747.

* Verified by repeated experiments.

Cylinder constant per explosion per lb. sq. inch...0·00709.
Pressure (above atmosphere) at end of compression
= 60 lbs. sq. inch.
Pressure maximum of explosion ....=168 lbs. sq. inch.

HEAT DISTRIBUTION IN 25 B.H.P. HORNSBY-AKROYD.

	Full load B.T.U.	Two-thirds load B.T.U.
Heat developed by combustion of oil per min. B.T.U. (oil used per minute 0·329 lbs. × 1900 B.T.U.)....	6284 = 100%	5329 = 100%
Heat turned into work on the piston B	1316 = 20·9%	....
Heat turned into work on the brake	1134 = 18%	14·3%
Heat carried away in jacket water	3172 = 50·4%	....
Heat in exhaust, &c. (by difference)	1796 = 28·7%	....

NOTE.—I.H.P. = 33000 ft. lb. per minute = 42·42 B.T.U. per minute.

The engine ran continuously, smoothly, and steadily, from 10-15 a.m. to 8-15 p.m., and the greatest variation in speed during the trial was exceedingly small, amounting only to about 0·5 per cent.

When engine stopped, all bearings were quite cool, none more than milk warm.

Throughout the trials the exhaust gases were mostly colourless.

The results of analysis, in a previous test, show that the exhaust gases consisted mainly of steam, carbonic acid, oxygen diluted with nitrogen, no traces of carbonic oxide being detected, so that this exhaust is not in any way objectionable. These products indicate that the oil is completely burned in the engine cylinder with an excess of air and oxygen.

TRIALS WITH 5 B.H.P. OIL ENGINE, NO. 3820, MARCH, 1899.

Condition.	Full load.	Half load.	Light.
Date, March 13th, 1899.			
Average speed	251 R.P.M.	252 R.P.M.	253 R.P.M.
Effective brake load	50 lbs.	25 lbs.	Nil.
Effective brake circumf....	13·4 lbs.	13·4 lbs.	13·4 lbs.
B.H.P.	5·096	2·558	Nil.
Russian oil per hour (lbs.)	4·0625	2·875	2
Oil per B.H.P. per hour...	·79 lbs.	1·12	...

Fig. 145 is an indicator diagram taken when the engine was running at 251 revolutions per minute, and the B.H.P. being 5·5.

The above trials were made with a good commercial engine; one of two that happened to be going through the testing department at the time. It was not specially prepared for test purposes.

Crossley Engine.—Fig. 146 shows a sectional elevation and plan of the Crossley vaporiser. The valve to the left of the plan view is held upon its seat by a spiral spring

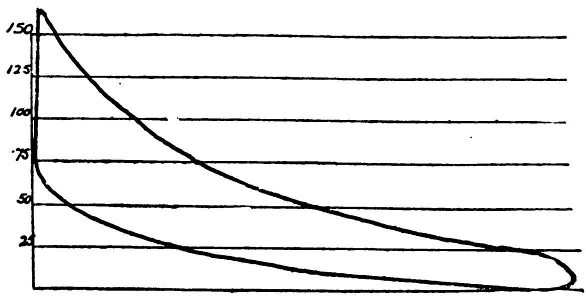


FIG. 145.

but opens communication to the cylinder during the outstroke of the piston. The governor acts upon this valve by a hit-and-miss arrangement, so that as the speed of the engine rises the valve is not opened. During the outstroke of the piston air follows through a separate automatic valve, held upon its seat by a spring; consequently there is a considerable suction through the passages connected to the valve shown when the latter is opened. Through these passages hot air and oil are simultaneously drawn, the oil entering at the pipe shown in the plan. The air enters the annular space round the chimney shown in elevation, and passes downwards to meet the oil.

A lamp is continually burning beneath the vaporiser, and,

besides heating the latter, its flame plays upon the ignition tube, which is connected to the combustion chamber of the engine. The products of combustion of the heating lamp pass up through the square holes shown in the vaporiser. In this way the heating surface is largely increased. The

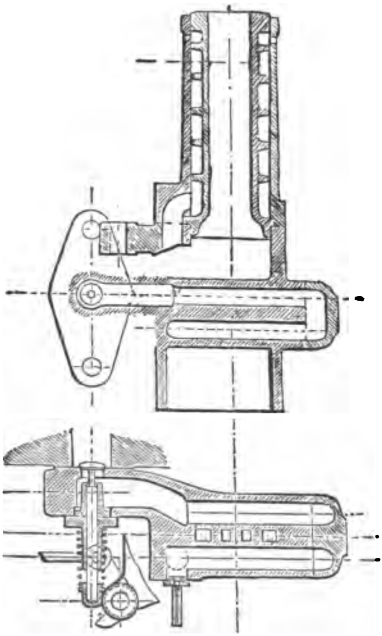


FIG. 146.—Sectional Elevation and Plan of Crossley's Vaporiser

average time taken to start the Crossley engine, at the Royal Agricultural Show trials at Cambridge, was, according to Professor Capper's report, 16 minutes. The engine ran without giving trouble, the governing was good, and the power developed was uniform throughout the trials.

The Wells Engine.—The oil feed to this engine is delivered by means of a rotating plug, driven by the link R, fig. 147

This plug allows a measured quantity of oil to drop on the inclined surface shown at V. Here it becomes vaporised by the high temperature maintained by an oil lamp and automatic air blast. The lever L is operated by a cam on the side shaft, and rocks upon a pin fixed in the boss E. The lower end of the lever opens the exhaust valve when pressed inwards against the tension of the spiral spring placed at its upper end. On the return stroke of the lever the oil plug is rotated, and oil emitted to the vaporiser at V; at the same time the air valve A is opened by the adjustable stud

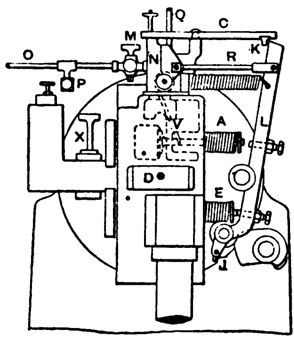


FIG 147.—Arrangement of Valves and Levers in Wells' Oil Engine.

on the lever L. By this ingenious arrangement all the valves are controlled by the one lever.

The governing is effected by the horizontal link C. This link is balanced so that its position of equilibrium is such that the upper end of the lever L clears the point K when the former is moving from right to left. When the upper end of the lever L is moved outwards by the cam, the horizontal lever C is drawn downwards; when released it recovers its position of equilibrium in a definite time, depending upon the spring adjustment at M. If the speed of the engine be increased so that L moves inwards before K has risen, then the further movement of the lever is arrested by the catch K. In this way the exhaust valve is kept open,

allowing the hot products of combustion to return into the cylinder; at the same time the oil and air valves are prevented from opening. The ignition of the charge is effected by means of a tube, heated by an auxiliary lamp.

Trusty Oil Engine.—Sectional views of this engine, which is made by Messrs. Weyman and Hitchcock, are shown at figs 148, and 150. and an end view at fig. 149. The oil supply is pumped, together with a little air, into the tube C, fig. 111, and in falling to the bottom of the annular space

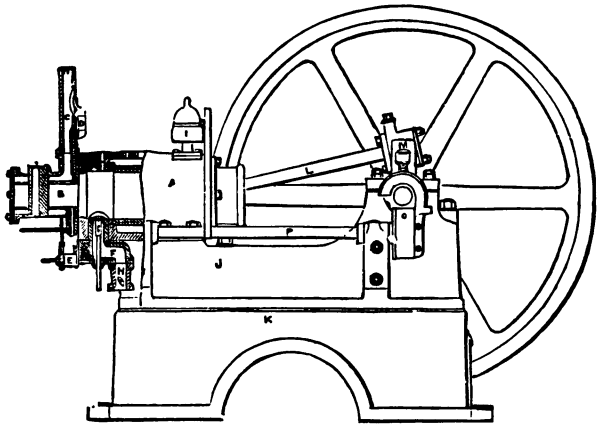


FIG. 148.—Section through Cylinder and Vaporiser of the Trusty Oil Engine.

becomes vaporised. The mixture then rises through the vapour valve, fig 150, into the inner chamber, and passes away from there into the combustion chamber. Here it is mixed with more air, which enters the cylinder through valve F, fig 148, on the outstroke of the piston. Ignition may be effected by contact with the hot chamber, though usually the ordinary hot tube igniter is used.

The following is an extract of a trial made by Mr. W. Worby Beaumont, M.Inst.C.E., on the Trusty engine, in 1893. The dimensions of the engine were as follow: Diameter of cylinder, 7·375 in.; length of stroke, 14 in.;

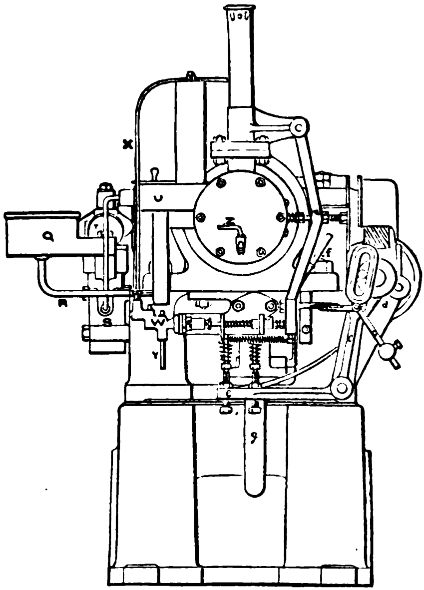


Fig. 149.—Arrangement of Valves, Levers, &c., in Weyman and Hitchcock's Trusty Oil Engine.

Fig. 150.—End Section of Trusty Vaporiser.

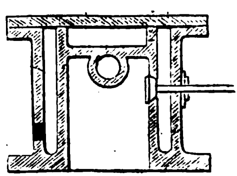


Fig. 151.—Sectional Plan of Trusty Vaporiser.

revolutions per minute, 248; flywheel diameter, 5 ft. According to Mr. Beaumont's report, the engine works equally well with all kinds of lamp oil.

"The duration of the several trials was: Full power, four hours; half power, two hours; running light, one hour. The intended full-power load was 6 horse power, and the brake load was adjusted as nearly as possible to this. The running of the engine was very regular, and when at half power the speed was within 0·4 of 1 per cent of that of full power. The oil used in the trials was Royal Daylight, costing in quantities about 4d. per gallon. Its specific gravity was found to be 0·802, one pint thus weighing 1·0025 lb."

The general results of the trials are as follow:—

	Full power.	Half power.	Running light.
Brake horse power	5·98	3·32	..
Indicated horse power	7·04	5·48	2·72
Mechanical efficiency	84 %	61 %	..
Oil used per B.H.P. hour ... lbs.	0·82	1·12	..
Oil used per I.H.P. hour ... lbs.	0·69	0·68	0·82

COST OF FUEL FOR TRUSTY OIL ENGINE WITH OIL AT FOURPENCE PER GALLON.

	Cost per I.H.P. hour.	Cost per B.H.P. hour.
Engine with full load	0·348d.	0·410d.
Engine with half load	0·342d.	0·565d.
Engine running light	0·406d.	....

The above figures do not include the cost of oil for heating the lamp. This is, however, small, and was found to be from 6 oz. to 7 oz. per hour. Thus, on the average, the total cost may be taken as not exceeding 0·4d. per I.H.P. hour.

CHAPTER XXV.

The Campbell Oil Engine.—The construction of the engine is shown in the engravings. It will be seen that the engine has only two valves—the inlet valve A (fig. 152) and the exhaust valve B (figs. 153 and 154). Each valve is fitted in a loose cone-shaped box ground into its seat. The valve can, therefore, be easily removed for cleaning, and as easily replaced. The valve A is held up by a spring, and opens

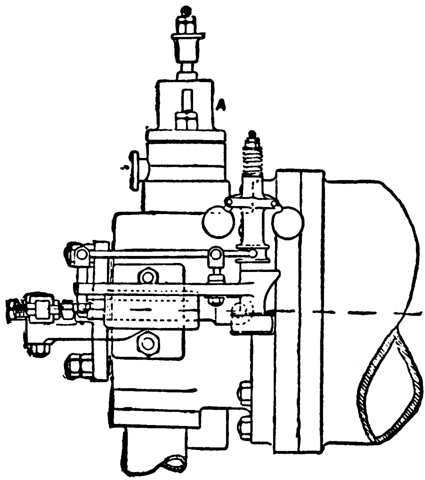


FIG. 152.

when a vacuum is formed in the cylinder. The valve B is worked through a lever and side rod by an eccentric on the crank shaft. When the speed exceeds the normal, a centrifugal governor pushes down a steel catch and prevents the exhaust valve closing. When this valve is held open, no vacuum can form in the cylinder during the suction stroke

of the piston, and consequently no charge of oil is drawn through the inlet valve A.

The oil is contained in the cistern above the cylinder. It flows by gravity through a pipe to a branch C in a circular chamber surrounding the inlet valve A on the top of the vaporiser. Small holes lead from the circular chamber to the conical face of the inlet valve. When the valve is drawn down by the vacuum created by the suction stroke of

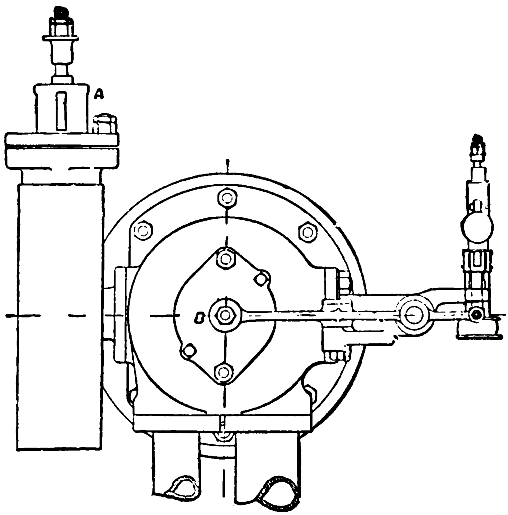


FIG 153.

piston, the oil can flow past the valve into the vaporiser; at the same time air is also drawn in, and spreads or sprays the oil against the heated sides of the vaporiser. The vaporised oil and the air go together into the cylinder and form the combustible mixed, which is compressed and then fired by the ignition tube. The vaporiser and ignition tube are kept hot by the same lamp, which is fed with oil by the pipe D.

Figs. 155 to 158 show sample diagrams taken during a

trial by Professor Stanfield. The following is quoted from the report referred to.

"The tests were similar in every respect to those carried out by me in connection with the Edinburgh Show of the Highland and Agricultural Society in July, 1899:

Date of tests	April 21, 1900.
Declared brake horse-power of engine...	13
Diameter of cylinder	9·5 in.
Stroke	18 "
Normal Speed revolutions per minute...	210
Description of oil used during tests.....	'Russolene.'
Specific gravity of oil	0·824

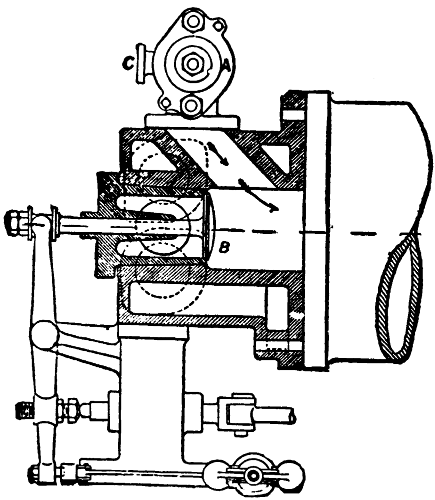


FIG. 154.

"The engine was run at full load for four hours without a stop; a trial lasting for two hours was then made at about half power; afterwards the engine was run light for one hour; finally a maximum power trial of half an hour's duration was made.

"The total revolutions and explosions during each test were taken by means of counters.

"The power was absorbed by a rope brake; the spring balance readings and all weights used were carefully checked at the conclusion of the tests.

"Indicator cards were taken at intervals of about fifteen minutes.

"The oil was contained in a cistern placed on the top of the engine cylinder, being fed by gravity to the vaporiser and vaporiser lamp.

"The oil cistern was fitted with a float gauge, by means of which it was possible to accurately determine the level of oil in the cistern at the beginning and end of each test. Oil was afterwards weighed in, and the amount thus ascertained.

"During a greater part of the full power trial the vaporiser lamp was not burning; the vaporiser being sufficiently heated to vaporise the oil and to bring about ignition at the end of the compression stroke.

"The following are the results of the several tests:—

Full Power Trial:

Duration of trial	4 hours.
Mean speed	210·26 revolutions per minute.
Effective circumference of brake...	16·029 ft.
Load on brake	156·5 lb.
Spring balance reading (average)...	10·06 lb.
Effective load on brake	146·44 lb.
Brake horse-power	14·95
Explosions per minute (average)...	80·17
Effective mean pressure	68·45 lb. per sq. in.
Indicated horse-power..................	17·68
Mechanical efficiency	84·50 per cent.
Oil consumption total..................	50·625 lb.
Oil per brake horse-power per hour	0·846 lb.
Oil per indicated „ „	0·715 lb.

"In connection with the above test, the consumption of oil and brake horse-power was ascertained at intervals during the run, the following particulars being obtained :—

Time.	Brake Horse-Power.	Oil Consumption per hour.	Oil Consumption per Brake H.P per hour.
		lb.	lb.
8-52½ to 9-52½	15·10	13·175	0·873
9-52½ „ 11-22½	15·00	12·770	0·847
11-22½ „ 11-47	15·08	12·500	0·830
11-47 „ 12-21	15·01	12·340	0·823
12-21 „ 12-52½	14·64	11·810	0·807

"The above results show a gradual diminution in the consumption of oil per brake horse-power per hour as the trial proceeded. At the beginning of this test too little cooling water was flowing through the cylinder jacket, consequently the cylinder became overheated : this fact no doubt accounts for the slightly heavier consumption during that time.

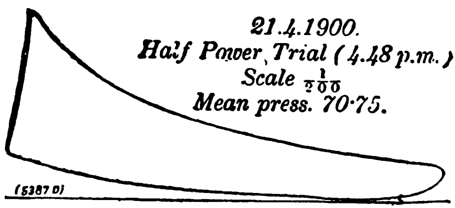


FIG. 155.

Half Power Trial :

Duration of trial	2 hours.
Load on brake	87 lb.
Spring balance reading, mean	4 lb.
Effective load on brake	83 lb.
Effective circumference of brake...	16·029 ft.
Revolutions per minute, average...	213·05
Brake horse-power	8·58
Explosions per minute, mean	48·24

Half Power Trial—continued.

Effective mean pressure	68·87 lb. per sq. in.
Indicated horse-power..................	10·70
Mechanical efficiency	80·20 per cent.
Oil consumption, total	16·44 lb.
Oil per brake horse-power per hour	0·957 lb.
Oil per indicated ,, ,,	0·768 lb.

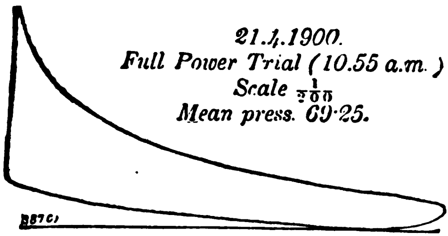


FIG. 156.

Light Power Trial:

Duration of trial	1 hour.
Revolutions per minute, mean	217
Explosions per minute, mean	17
Effective mean pressure	54·55 lb. per sq. in.
Indicated horse-power	2·98
Oil consumption, total..................	3·313 lb.
Oil per indicated horse-power per hour	1·111 lb.

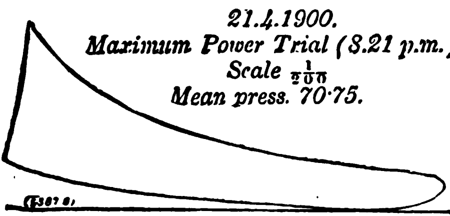


FIG. 157.

Maximum Power Trial:

Duration of trial	0·50 hour.
Load on brake	192 lb.
Spring balance reading, mean	15 lb.
Effective load on brake	177 lb.
Revolutions per minute, mean......	207·8
Effective circumference of brake...	16·029 ft.
Brake horse-power	17·86
Explosions per minute, mean	93·2
Effective mean pressure	66 83 lb. per sq. in.
Indicated horse-power..................	20 06
Mechanical efficiency	89 00 per cent.
Oil consumption, total..................	6·9 lb.
Oil per brake horse-power per hour	0·773 lb.
Oil per indicated " "	0·687 lb.

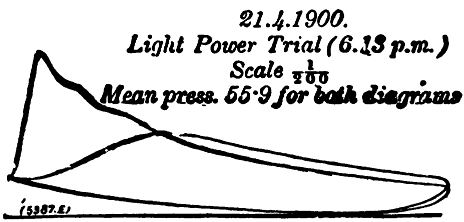


FIG. 158.

Summary of the above trials.

Conditions of Trial.	Full Power.	Half Power.	Light Power.	Maximum Power.
Brake horse-power	14·95	8·58	..	17·86
Indicated horse-power.........	17·68	10·70	2 98	20·06
Mechanical efficiency	84·50	80·20	..	89·00
Oil consumption, hour lb.	12·656	8·22	3·313	13·8
Oil consumption per brake horse-power per hour lb.....	0·846	0·957	..	0·773
Oil consumption per indicated horse-power per hour lb.....	0·715	0·768	1·111	0·687

"The engine appeared to work exceedingly well, and it is to be regretted that time did not permit me to continue the maximum power trial, as I am convinced the engine could have maintained that load for a considerable time without showing any signs of distress.

"All bearings were quite cool at the end of the tests, and the combustion of the oil seemed to be perfect as the exhaust was quite clear."

The author was asked to make a further test of a Campbell engine with oil of a somewhat different character. The following is quoted from the author's report :—

"The vaporiser fitted to this engine was made according to the pattern usually supplied by the Company, and known as the 'Campbell' Vaporiser. The oil was fed by gravity to the vaporiser and to the lamp for heating the same.

"The speed of the engine was regulated by means of a centrifugal governor, by the action of which the exhaust valve was caused to remain open when the speed increased slightly beyond the normal. A tachometer showed a maximum deviation during the tests of 2½ per cent from the mean speed.

"The engine is extremely simple in its construction, there being only one valve for the admission of the oil and air charge to the cylinder, and one other valve for the exit of the burnt gases. Both these valves can be detached from the cylinder in a few moments for examination.

"The ignition of the charges was obtained by means of a hot tube attached to the vaporiser, and heated in the first instance by means of a petroleum lamp. When running with the full load on the engine the lamp may be dispensed with.

"The time required for heating the vaporiser from normal atmospheric temperature to that necessary for starting the engine was fourteen minutes.

"Referring to the tabulated figures, it will be seen from line 14 that the percentage of carbon in the petroleum is less than usual. This figure has been checked by an independent analysis, and is thereby confirmed.

"The exhaust gases were analysed by means of an Orsat apparatus, the gases being collected from a branch on the exhaust near the engine. The steam in the exhaust gases was of course condensed in the collecting pipes. Each sample was tested for combustible gas, but none was detected, thus proving the combustion to have been complete.

"It is sometimes maintained that engines working with a gravity feed for the oil do not supply oil uniformly to the vaporiser. I found, however, that the constituents of the exhaust gases bore nearly the same proportion on the six samples tested, thus showing that the oil supply was very regular, and that no leakage occurred at the oil inlet valve.

"Referring to line 19, the temperature of the exhaust gases was probably higher than that calculated from the indicator diagram. Although the 'pellet' of the Siemens Pyrometer was inserted in the flame from the exhaust, it was difficult to plunge it into the water in the instrument without some loss of heat. This accounts for the discrepancy between the figures given.

"Line 23. This gives the thermal efficiency of the engine to be 16·4 and 14·5 at full and half loads respectively. The efficiency is here calculated on the brake horse power, not on the indicated, as is sometimes done. Had this been done the figures would be 20·3 per cent at full load and 21·1 per cent at half load. It is obvious that a greater proportion of heat is absorbed in driving the engine at half load than at full load, which accounts for the variation of the figures.

"Throughout the tests the engine worked smoothly and without attention. The exhaust gases were colourless, and upon holding a sheet of white paper to an orifice near the engine, it did not become blackened by particles of unburnt carbon."

TABULATED RESULTS OF TEST.

1. Nominal dimensions of engine—9½in. diameter cylinder and 1ft. 6in. stroke.
2. Exact area of piston 71·54 square inches, and 1·49 feet stroke.

	Full Load.	Half-Load.	Light.
3. Revolutions per minute	211·55	215·05	216·67
4. Deviation from mean speed by tachometer	2½%	2½%	2½%
5. Net load on brake in pounds	168·73	90·95	——
6. Brake horse power........................	17·37	9·52	——
7. Indicated horse power	21·59	13·7	4·2
8. Mechanical efficiency per cent...	80·4	69·3	——
9. Mean effective pressure in pounds per square inch......................	72·5	71·9	54·5
10. Mean number of explosions per min.	92	59	24
11. Compression pressure in pounds per square inch......	50	48	30

12. *Oil.* Specific gravity at 70 deg. F., 0·793.
13. Flash point (barometer 30 in.), Abel close test, 84 deg. F.
14. Percentage of carbon by weight, 78·89.
15. Percentage of hydrogen by weight, 13·88.
16. Calorific value per pound of oil, deducting for latent heat of steam formed, 18,847 B.T.U.
17. Exhaust gases. Percentage of constituents by volume .... CO_2 11·4 per cent. O 3·3 per cent. N 85·3 per cent.
18. Specific heat of exhaust gases, allowing for steam formed by combustion of hydrogen, 0·177.
19. Temperature of exhaust gases calculated from indicator diagram, 1270 deg. F. Temperature taken by Siemen's pyrometer, 1014 deg. F.

	Full Load.	Half-Load.	Light.
20. Jacket water per minute—pounds ...	24·4	15·7	5·15
21. Temperature of jacket water inlet...	44	44	44
22. Do. do. outlet...	117	119	112
23. Percentage of heat energy available at flywheels	16·4	14·5	——
24. Percentage of heat absorbed in driving engine	3·9	6·4	14·1
25. Percentage of heat lost in jackets ...	39·9	42·4	27·9
26. Percentage of heat lost in exhaust gas and by radiation	39·8	36·7	58·0
27. Oil consumption. Total used, including lamp—pints	42·9	17·85	4·04
Do. pounds	42·48	17·67	4·00
28. Oil per I.H.P. hour—pints	0·662	0·651	0·957
Do. do. pounds	0·656	0·643	0·948
29. Oil per B.H.P. hour—pints...........	0·823	0·937	——
Do. do. pounds.........	0·815	0·928	——

The Britannia Company's Engine is the invention of Mr. Roots. Fig. 159 shows a section through the vaporiser of the Britannia oil engine made under Roots' patents. I is the ignition tube, which is surrounded by the casing H. Air enters the casing, and in passing round the spiral passages it becomes heated by the same lamp used to heat

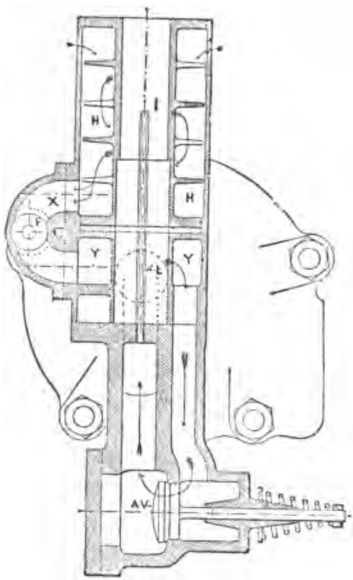


FIG. 159.—Section through the Vaporiser of the Britannia Oil Engine. The arrows show the direction of flow of the air.

the ignition tube. During the outstroke of the piston the air is sucked from H, through the elbow at X, to Y, and so passes into the cylinder through the valve A V. As the air sweeps through the passages marked X, it carries away with it a certain quantity of oil from the grooves of a spindle

which enters the chamber X at the point marked F. The oil is vaporised by contact with the hot walls of the chamber Y, and passes off with the air to the cylinder. On the return stroke of the piston the vapour is compressed into the ignition tube and ignited.

The way in which the oil is carried into the chamber, shown at X, fig. 159, is further illustrated in fig. 160. The spindle is here shown at A A, and the side view of the chamber X shows the spindle A A passing through it. This

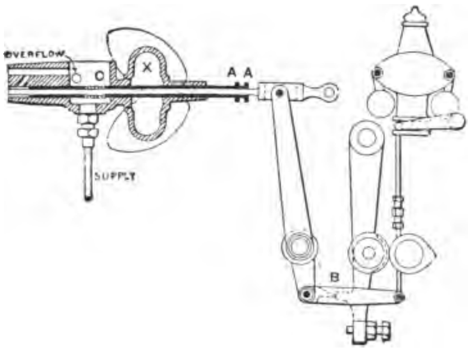


Fig. 160.—Arrangement of Oil Supply and Governor Gear of the Britannia Oil Engine.

spindle has grooves cut upon it, which are shown in the oil cavity C, fig. 160. The quantity of oil entering the chamber X is regulated by the governor. As the speed rises the piece B is lifted, so that all the grooves on the spindle do not enter the chamber X; thus the oil supply is reduced.

According to Professor Capper's report, from which we have produced the illustration, this engine gave no trouble in working. The average revolutions on each of the three days of the Cambridge trials did not vary, though there was some racing of the engine in spite of the ingenious method of governing.

The Capitaine Oil Engine is made by Messrs. Tolch and Co., of London, and its distinguishing features are shown in figs. 161 and 162. Fig. 161 shows the upper end of the vertical combustion chamber. On the outstroke of the piston air enters the automatic valve shown at B. The greater volume of air passes into the cylinder round the outside of the case D, whilst some air necessarily passes through the conical hole C. The part C is kept at a high temperature by the explosion, its heat being retained by the non-conducting substance surrounding it.

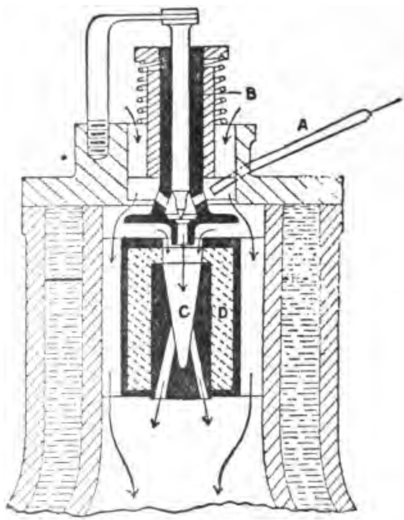


FIG. 161.—Section through Vaporiser and Combustion Chamber of Tolch and Co.'s Capitaine Engine.

Oil enters by the pipe A, and, passing through the hole in the air valve B, drops into C, and is vaporised. The central spindle within the valve B closes the oil passage to C, when the valve is in its uppermost position. The governing is effected by holding open the exhaust valve, so that B

remains closed during the outstroke, and no air or oil is passed into the cylinder.

The delivery of oil to the pipe A is effected by the plunger pump B, shown in fig. 162. This pump works in a glycerine bath. Oil floats upon the surface of the glycerine and passes into the pipe A, fig. 161 through the slide valve A, fig. 162.

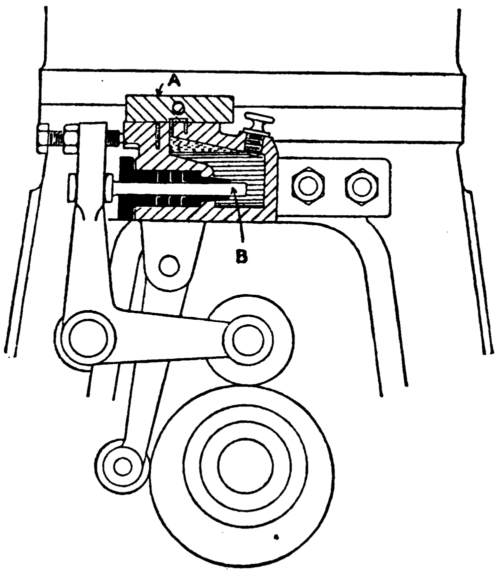


FIG. 162.—Glycerine Pump of Tolch and Co.'s Capitaine Engine.
A—Slide Valve. B—Pump Plunger.

Professor Capper says: "The oil ordinarily used in this engine is Tea Rose, and some difficulty was experienced in keeping the vaporiser hot enough to work with Russolene. Experience with the use of Russolene oil may be expected to overcome these difficulties, and the engine certainly deserves praise for its particularly quiet running, and for the ingenuity as well as simplicity of its working parts."

Tangye's vaporiser, made under Pinkney's patents, is shown in fig. 163. The vaporising chamber is in direct communication with the cylinder. Air is drawn in through the mushroom valve, and meets oil fed by gravity from a tank to the hole on the valve seating. Thus, when the valve opens by the suction of the piston, oil drops through into the vaporiser. To govern the engine the exhaust valve

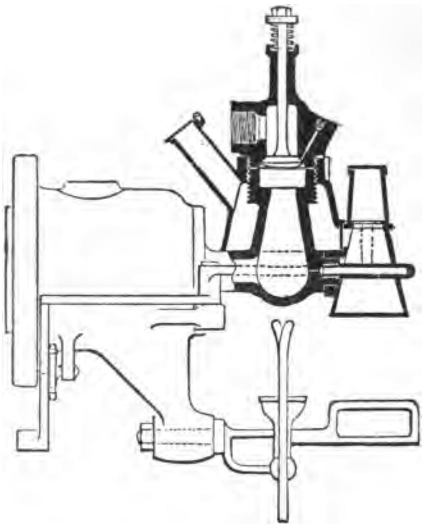


Fig. 163.—Tangye's Vaporiser.

is kept open and the automatic valve remains closed during the suction stroke, thus preventing the admission of fresh air or oil to the cylinder. A lamp placed beneath the ignition tube heats both the vaporiser and the tube. Before starting, the lamp is placed immediately beneath the vaporiser.

The Fielding Oil Engine is constructed by Messrs. Fielding and Platt Limited, of Gloucester. The engine works in a similar way to a gas engine, with the exception of the additional parts required to vaporise the oil. A longitudinal

section of the vaporiser is shown in fig. 164. This vaporiser consists of two tubes, one above the other, the lower one being heated by the lamp to a bright red, whilst the upper one is kept at a lower temperature. Two copper tubes, at a high temperature, connect the air inlet shown to the upper vaporiser tube. V is an automatic valve, drawn open by the outstroke of the engine piston. The action of the engine

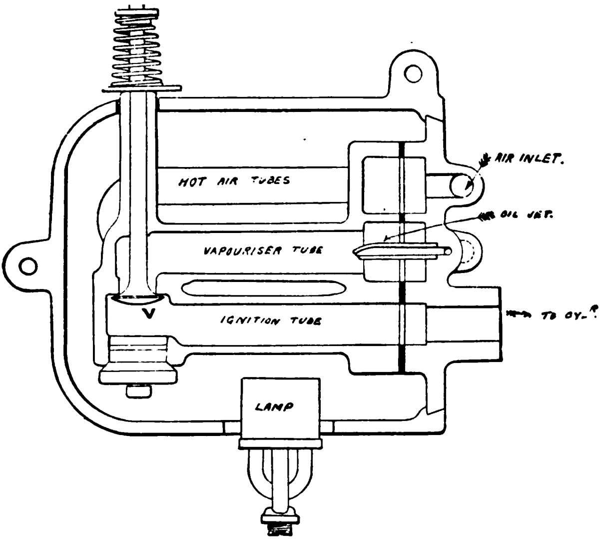


FIG. 164.—The Fielding Vaporiser.

is as follows: During the outstroke of the engine piston, air is drawn into the cylinder through a valve in the combustion chamber (not shown). The valve V also opens, and a current of hot air sweeps through the ignition tube into the cylinder. Oil vapour, which has been pumped into the vaporiser tube mixes with this air, but is not ignited by the ignition tube because of the inferior quantity of air drawn through the valve V. When the vapour charge enters the cylinder it

mingles with the pure air entering by another valve, and thus forms an explosive mixture, which ignites when part of it is compressed back into the ignition tube. Fig. 165 shows an end view of the engine.

The speed is controlled by a centrifugal governor, which props up the exhaust valve lever and the oil pump rod. Hence no oil is injected into the vaporiser, nor is pure air

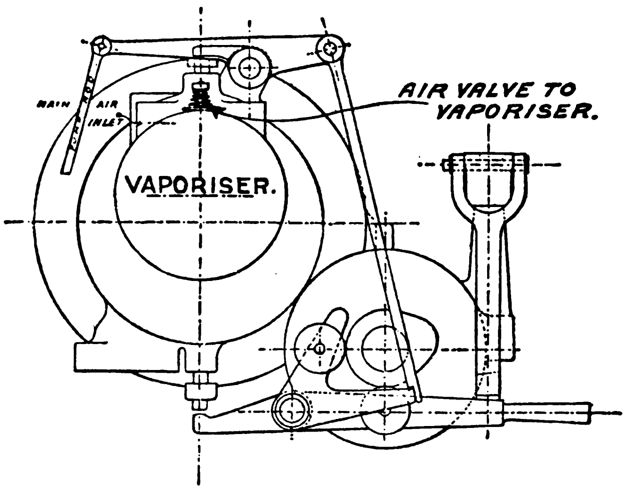


Fig. 165.—Elevation showing Arrangement of Valve Levers and Trip Gear of Fielding and Platt's Oil Engine.

drawn into the cylinder through either of the valves. This engine ran very steadily during the Cambridge trials, starting readily with one attendant after the vaporiser tubes were heated for about 20 minutes. This engine was, however, withdrawn from the competitive trials on account of slight defects in the heating lamp, which have since been remedied.

A section of Messrs. Clayton and Shuttleworth's method of igniting is shown in fig. 166. A steel needle projects into

the combustion chamber. This needle is surrounded by a number of strips of asbestos millboard. The heat retained by this arrangement keeps the needle at a sufficiently high temperature to ignite the charge when it is compressed. A tube shown in the illustration serves for ignition when starting the engine. The igniter is said to work satisfactorily even at low loads.

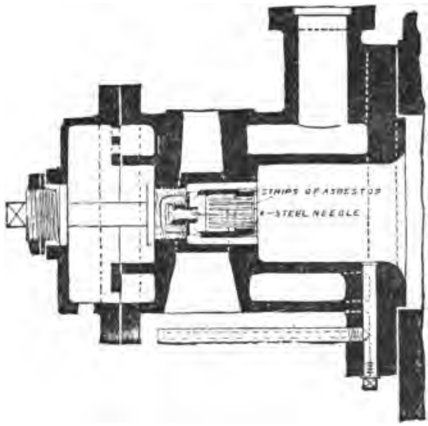


FIG. 166.—Section of Clayton and Shuttleworth's Oil Engine Igniter.

The Gardner Oil Engine (figs. 167, 168, 169).—This engine is manufactured by Messrs. L. Gardner & Sons, of Barton Hall Engine Works, Patricroft. The general outline of the engine is similar to that of the gas engine already described as made by this firm. The eccentrics for operating the valves are similar to those used on the gas engine, and constitute a distinguishing feature of the engine. The governor also acts upon the same principle as that described for the gas engine; such structural modifications being made to enable the oil pump to be operated only when the vapour valve is opened. Thus, as in the case of the gas engine, the governing is effected by cutting out the supply

of oil to the vaporiser, and at the same time preventing the vapour already in the vaporiser from entering the cylinder.

The oil pump for forcing the oil to the vaporiser is shown on the general drawing, figs. 167, 168, and also in detail in

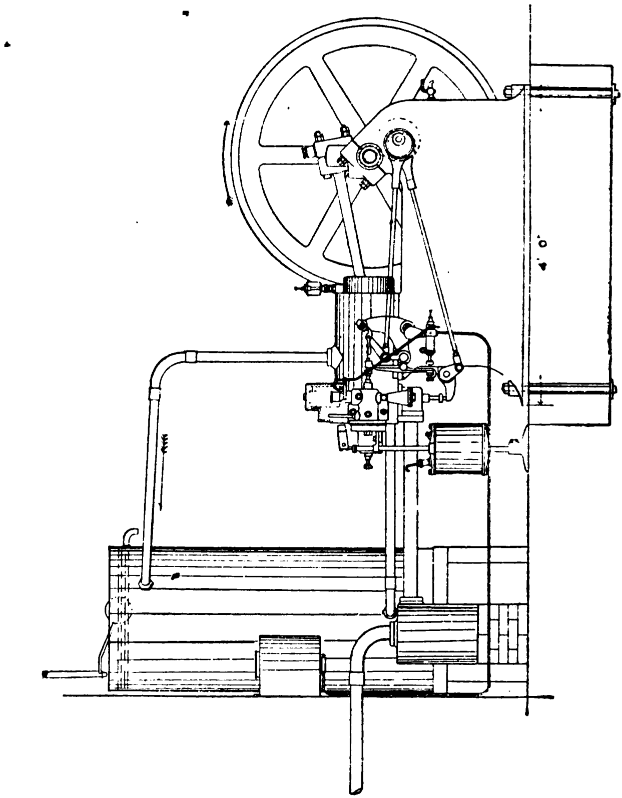


FIG. 167.

fig. 169. The plunger P is forced inwards by a rocking lever shown clearly in the general drawing. As the plunger moves forward it cuts off the oil inlet port through which

THE GARDNER OIL ENGINE.

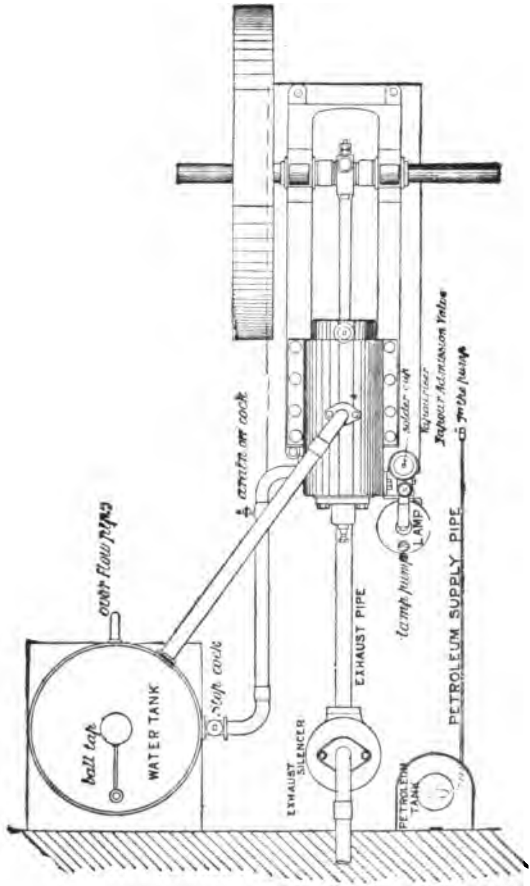


FIG. 168.

oil enters the barrel of the pump by gravity from the supply tank. The oil charge is then forced past the foot valve and delivered to an open cup on the vaporiser, whence it is

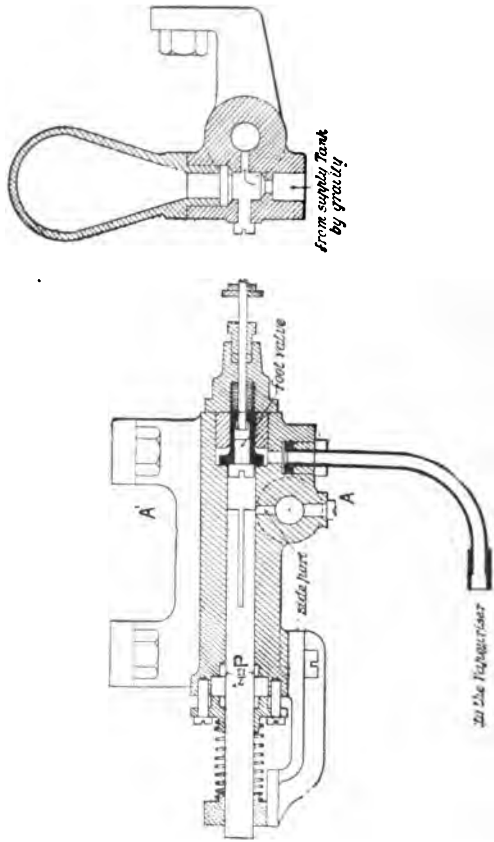


Fig. 169.

sucked into the vaporiser during the indraft caused by the piston. A little air enters the vaporiser with the oil, but the greater supply is taken through the air regulator bolted

to the admission valve box. From the positions of the two channels, it will be seen that when the admission valve opens to the cylinder, a stream of vapour and air meet, and become well mixed as they pass together through the valve to the cylinder. Ignition is effected by the hot tube heated by the same lamp used for the vaporiser.

The following test figures for these oil engines have been supplied by the makers.

No. 1 Oil Engine—

1¼ B.H.P.
Cylinder diameter, 3½ in.
Piston stroke, 5 in.
Oil per B.H.P. hour (including lamp), 1·13 lb.
Petroleum used, 0·825 density.
Oil used per B.H.P. hour on larger engines, 0·7 lb.

The Diesel Oil Engine.—Each advance made by designers in reducing the losses in gas and oil engines has been obtained by increasing the amount of compression of the charge before ignition, until to-day a pressure of 80 lb. is often met with. It is probable, however, that in the usual form of engine the limit is practically reached, for the high temperature of the piston, etc., usually found associated with a charge of highly-compressed and combustible gas is running very close to the point of explosion. Herr Diesel's aim has been to increase the pressure due to compression far beyond that mentioned above, and to do this successfully he has adopted a modified cycle, in which the air is compressed separately and the oil injected afterwards, the ignition being effected by the temperature of the compressed air, which it was stated was about 1,000 deg. Fah. The oil burns, but not rapidly enough to raise the pressure in the cylinder, and the diagrams obtained when running under full load resemble somewhat those obtained from a steam engine, as the upper line is very nearly horizontal until the oil supply is cut off, when the expansion curve commences. The large excess of

air present serves to keep the temperature of the cylinder within reasonable limits, and ensures the complete combustion of the charge.

The engine is operated on the two-cycle system, the operation when working being as follows :—When about 90 per cent of the working stroke has been completed the exhaust valve opens, and the air inlet valve admits air at a pressure of 4 lb., which sweeps out the cylinder, clearing away the products of combustion and filling the cylinder with fresh air. The exhaust valve is then closed, and the piston compresses the air into a very small compression space, ready for the injection of the oil spray at the commencement of the out or working stroke of the piston.

A recent engine of this type, built by Messrs. Scott and Hodgson, was of the horizontal type, the piston being $7\frac{7}{8}$ in. diameter and $10\frac{3}{4}$ in. stroke, and was run at 216 revolutions per minute. On the piston rod, between the power piston and the crank, is a second piston 9 in. diameter working in a cylinder, which fulfils two functions. During the first part of the instroke it compresses air, and delivers it at a pressure of 4 lb. per square inch into a reservoir formed in the engine bed, for the purpose of cleaning out from the power cylinder the products of combustion, as mentioned above. The discharge valve is over-run by the piston before the stroke is completed, and the air remaining in the power cylinder is further compressed to 60lb. per square inch, and delivered to a second pump $2\frac{3}{4}$ in. in diameter and $5\frac{1}{8}$ in. stroke, and by which the compression of the air is carried on ; a pressure of 750 lb. per square inch is attained and is used to spray the oil into the power cylinder. The oil pump is operated by an eccentric driven by a lay shaft at the side of the engine, and the governor operates by means of a taper wedge, which controls the suction valve, thus permitting some of the oil to leak back when the engine is running on light load. The pump delivers the oil to the inlet valve, and when the valve is opened the compressed air from the small pump forces it

along a number of fine passages into the power cylinder in the form of spray.

The surplus compressed air from the small pump is stored in a steel reservoir for use in starting the engine. The operation of starting is effected by shifting a cam on the lay shaft, so that the oil inlet valve is put out of action, and another special valve is set in motion, which admits the compressed air from the two reservoirs, so that the engine is operated virtually as a compressed air engine until the engine is fairly in motion, when a small movement of the hand wheel, close to the end of the cylinder, cuts the air supply out and admits the oil supply.

The only tests available are some made abroad, and Professor Unwin states, as the results of tests and experiments that he has made upon an engine at Augsburg, that it will convert 37 per cent of the heat in the oil into useful work, which is a very considerable advance upon existing records of other oil engines. The Augsburg Engineering Company guarantee their customers a consumption of 0·452 lb. of solar oil or of crude petroleum per brake horse power hour, and that at half power the consumption shall not exceed 0·52 lb. for all sizes of engine. Professor E. Mayer, of Charlottenburg, reports a consumption at full power of 0·467, at three-quarter load 0·488, at half load 0·567, using raw Legernsee oil of specific gravity 789 at 68 deg. Fah. With oil at 4½d. per gallon, the Diesel engine is guaranteed to give at half load a brake horse power at the low cost of 0·26d. per hour. At work this engine ran very quietly, and with the advantage of operating on the two-cycle method the flywheel power required for any degree of steadiness is much reduced; whilst the system of governing is much more conducive to steadiness than the hit-and-miss method can possibly be. The complete combustion renders the deposition of soot in the cylinders and passages an impossibility.

CHAPTER XXVI.

Oil Gas Engines.

In writing upon the subject of oil and spirit engines as distinct from the parent motor, the gas engine, the progress made in connection with the latter is apt to be lost sight of. For instance, Mr. Daimler attacked the problem of obtaining motive power from oil or spirit, after having acquired a considerable knowledge of the peculiarities of internal combustion engines from his association with Dr. Otto in helping him to perfect the gas engine. Mr. Daimler took up the investigation of petroleum motors at a time when the gas engine had acquired a commercial reputation, and was, by far-sighted engineers, regarded as a serious competitor to small steam engines. The problem of the oil gas motor was by no means of easy solution, but Daimler, being already in a unique position as an inventor of internal combustion engines, was well qualified to design an engine working by gas generated from oil or spirit by the engine itself. Just as Otto had designed a successful gas engine from the raw material of many crude and impracticable patents, so Daimler may be regarded as the first designer of those engines which derive their power from petroleum spirit, and which were first applied to the propulsion of small vehicles on common roads. In saying this we have no intention of discounting the excellent work of Messrs. Priestman, who so perseveringly worked out the patents of Eteve; for there is a broad mark of distinction between all petroleum spirit engines and those which work by means of the heavier commercial petroleum so universally used as an illuminating agent. We are, however, prepared to admit, and, indeed, emphasise the great assistance which any inventor obtains from those commercial agencies by which his name becomes permanently associated

with industry. Indeed, in reviewing the vast mass of information contained in the annals of the Patent Office, we are almost forced to the conclusion that the inventor who succeeds owes more to his own business capacity, or that of his financiers, than he does to his professional attainments.

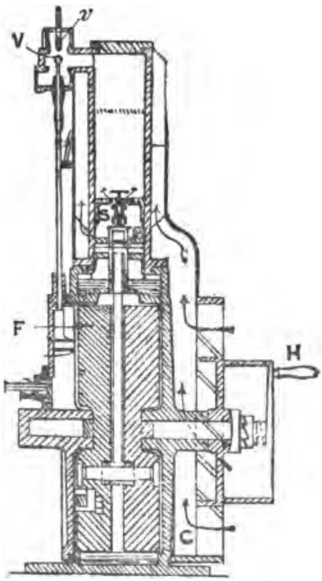


FIG. 170.

After several years' experience with the Otto gas engine, during which time Mr. Daimler assisted Messrs. Crossley Brothers, of Manchester, he devoted his attention to the perfection of a motor worked by petroleum spirit. The result of his earlier labours is embodied in the engine illustrated in fig. 170, which was patented in 1885. About a year previously to this Mr. Daimler took out a patent for tube ignition in connection with oil engines. Whether this

was a valid patent, however, may be doubted, for tube ignition in gas engines had been used by Mr. Atkinson as early as 1879. As, however, the validity of contested patents so often hangs upon threads worn thin by legal argument, it is possible that the application of the hot tube to oil engines was a sufficient departure from the usages of the times to constitute a useful patent. Referring to fig. 170, this motor worked upon the Otto cycle—that is, it gave an impulse to the piston every other revolution. The fly-wheel F, carrying the crank pin, is enclosed in the chamber C, open to the lower end of the cylinder. As the piston approaches the lower end of the cylinder, after ignition has taken place, the exhaust valve V is opened, and allows some of the products of combustion to escape. At the same time, a projection compresses the spring S, and leaves the valve in the piston free to admit air or combustible mixture to the cylinder; thus, at the end of the down stroke, some of the burnt products are displaced by the compressed gases in the chamber C, and, on the up stroke, more products are expelled from the cylinder. During the next down stroke the charge of combustible gas is drawn in through the valve V. When the bottom is again reached, the valve V is freed by the projection, and a further charge admitted from the chamber C; on the up stroke compression takes place, and then ignition by a hot tube. A previous ignition was said to be avoided by closing the gas valve towards the end of down stroke by a tappet. In the light of present knowledge, it would appear that this precaution was unnecessary, the ignition tube being filled with exhaust products would not permit the entry of an explosive mixture, unless the latter were considerably compressed within the ignition tube. The exhaust valve V is actuated by a cam on the flywheel through a pin and rod.

The engine was started by keeping open the exhaust and inlet valves by means provided, and turning the handle H. A clutch gear will be noticed attached to the starting handle to allow the engine to overrun the motion of the

handle when the first impulse occurred. The main object of the central valve in the piston was to accelerate the discharge of the exhaust, and to clear, as far as possible, all waste gases from the clearance space. That this is done at some risk of losing, through the exhaust, a portion of the new mixture, will be evident. It is stated, however, that the cycle might be varied by holding open the inlet and exhaust valves at suitable portions of the stroke, so that the valve in the piston could not come into operation. It is probable that, when the central valve in the piston was used, it was intended only to admit air from the lower chamber, in which case the air would greatly assist in clearing the cylinder of the exhaust products of combustion.

In 1885 Mr. Daimler proposed and patented the application of this motor to a bicycle. Previously to this date compressed air and steam had been tried in small motors adapted to cycles, as they then existed, but with little success. Indeed, compressed air at that time was impracticable ; compressors were scarce, as well as inefficient, and even if they had been available a convenient storage was wanting. Steam naturally attracted the attention, but the cumbersome boiler, the troubles of stoking, the space required, and the weight of the machinery, were obstacles sufficient to prevent the most sanguine inventor from contemplating their use as a means of propelling vehicles suitable only for one or two persons. With the spirit engine these disadvantages are considerably reduced ; weight is minimised, the fuel supply is not bulky, nor difficult to replenish, the motor small, and the working parts few. The general drawing which accompanied the patent is here reproduced (fig. 171). It may be regarded with contempt by the fastidious cyclist of to-day. But it must be remembered that, when this drawing was made, it was necessary to distinguish the "spider" bicycle from its dying contemporary, "the boneshaker." Some modifications in the gas generator were introduced. The vapour was generated in a vessel charged with petro-

leum, and containing a float fitted with a central basin, into which the petroleum rose through a hole in the bottom. A perforated air supply pipe led into this basin, and the air bubbling through the petroleum became charged with the vapour, and was led to the engine, a piece of wire gauze and a safety valve being inserted in its way to prevent accident in case the flame should strike back. The cylinder was cooled by a current of air generated by a fan on the crank-shaft. A rope running on grooved pulleys, and capable of being tightened by a jockey pulley, served to transmit the motion of the engine to the rear driving wheel. It may be claimed that the success attending this experimental machine

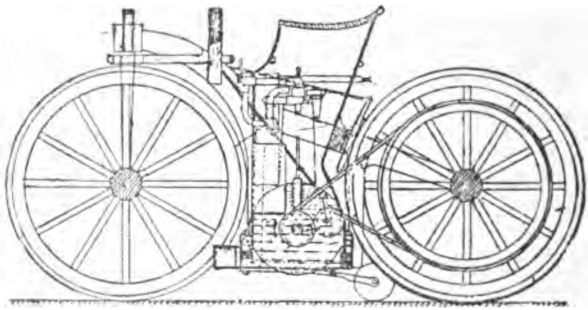


FIG. 171.

paved the way to the foundation of that industry which is now established, and is growing very rapidly.

In a short time the Daimler motor was used for propelling tram cars, carriages, quadricycles, launches, and was even fitted to fire engines. In 1893 the London County Council ordered a pinnace fitted with a Daimler motor, for use in connection with the sewage outfall works at Crossness and Barking. The following description appeared in the *Engineer:* We illustrate the motor and vaporiser then used. It will at once be noticed that the engine has undergone alterations. The most obvious improvement is the cam

action which replaces the grooved cam in the flywheel (fig. 172) for operating the exhaust valve. In this case the exhaust valve is actuated by a cam driven by two-to-one gearing from the crankshaft. The inlet valve is entirely automatic, and the central valve in the piston is dispensed

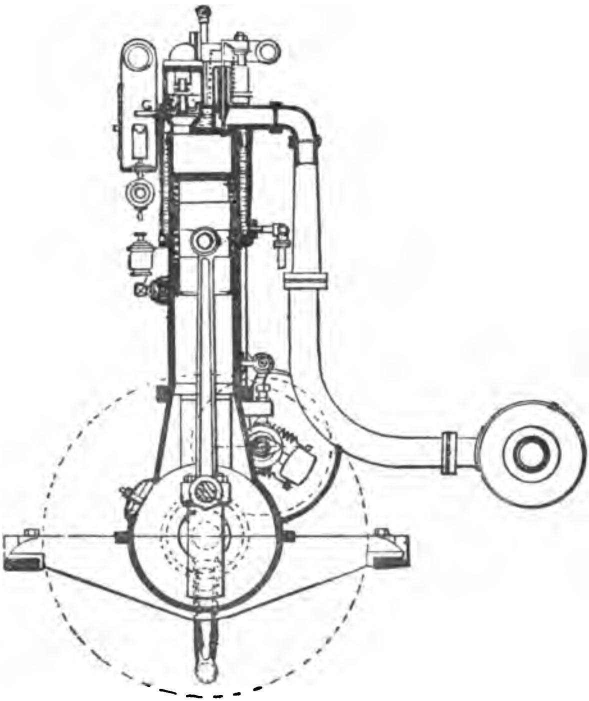


Fig. 172.

with. The engine governs by holding open the exhaust valve, as in the previous design, though the governing mechanism is much improved by the adoption of a weight, the centrifugal action of which trips the lifting finger of the

exhaust valve. The arrangement of the vaporiser is clearly seen by reference to fig. 173. The supply pipe S conducts the spirit (sp. gr. 0·68 to 0·70) to the bottom of a reservoir

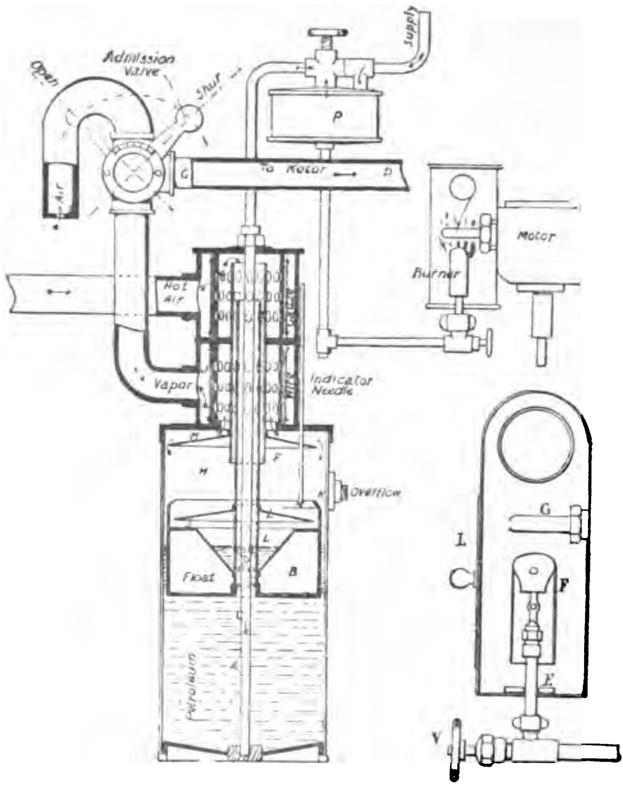


FIG. 173.

R, in which the spirit rises to about two-thirds of its height. The float F rests upon the surface of the liquid in the reservoir. The pipe P serves the double purpose of guiding

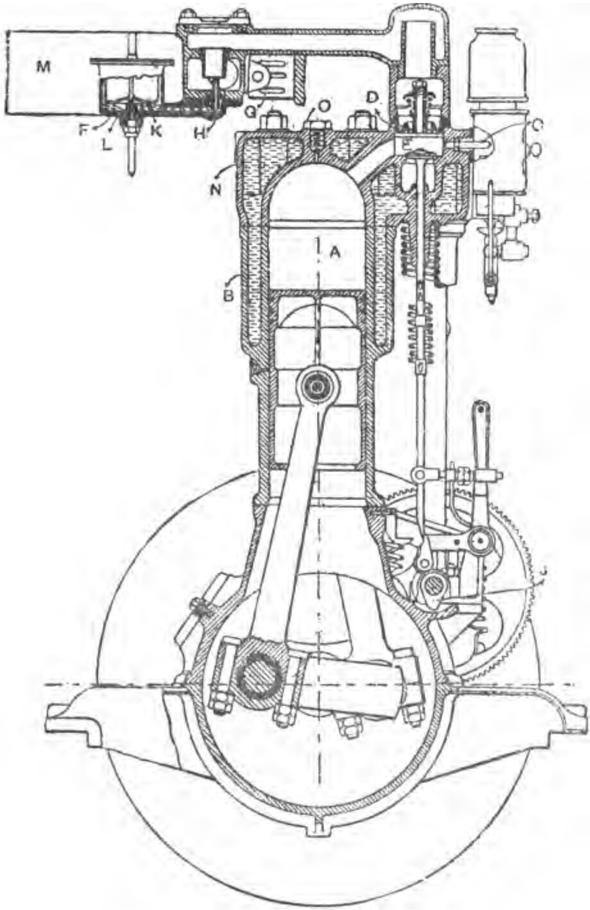


FIG. 174.

the float as it rises and falls and of carrying the air to the surface of the liquid, allowing it to bubble through the perforations shown at H. It will be seen that the main object of the float is to keep the pipe P with its perforations at the surface of the liquid as the latter rises and falls in the reservoir. In addition to this, the float prevents the agitation of the surface of the liquid, and ensures the uniform carburation of the air by keeping the perforations at the lower end of P immersed at a constant depth. We shall see in another design how the float is discarded for the purposes here mentioned, but is still used for controlling the supply of spirit. Hot air from the pipe A passes down the central pipe P, bubbles through the holes at H, and rises into the vapour pipe V. At G the air is strained through fine wire gauze. This filters it, and prevents accident due to the firing back of the motor should the flame so created reach the pipe V. It will be seen that an additional air supply is mixed with the vapour on its way to the motor. The air valve permits of the regulation of the air in the motor to suit the conditions of working. Change in temperature, and the humidity of the atmosphere, will affect the quantity of air required.

Since 1893 the Daimler motor has undergone still further development. We will now describe the engine as it is at present manufactured by the Daimler Motor Co. Ltd., of Coventry, to whom we are indebted for the illustrations herewith. Figure 174 shows a sectional elevation of the motor. The float F is now used as a regulator for the supply of spirit to the nozzle H. The petrol is forced up the supply pipe L by air pressure maintained during running by allowing some of the exhaust gases to leak through into the petrol chamber. On the down stroke of the motor piston A, the inlet valve E is opened against the pressure of a spiral spring. A current of air entering the adjustable slots shown at G passes into the channel just above the nozzle H. The induced draft of air carries with it a spray of petrol from H to the cylinder A. On the return stroke of the piston, the

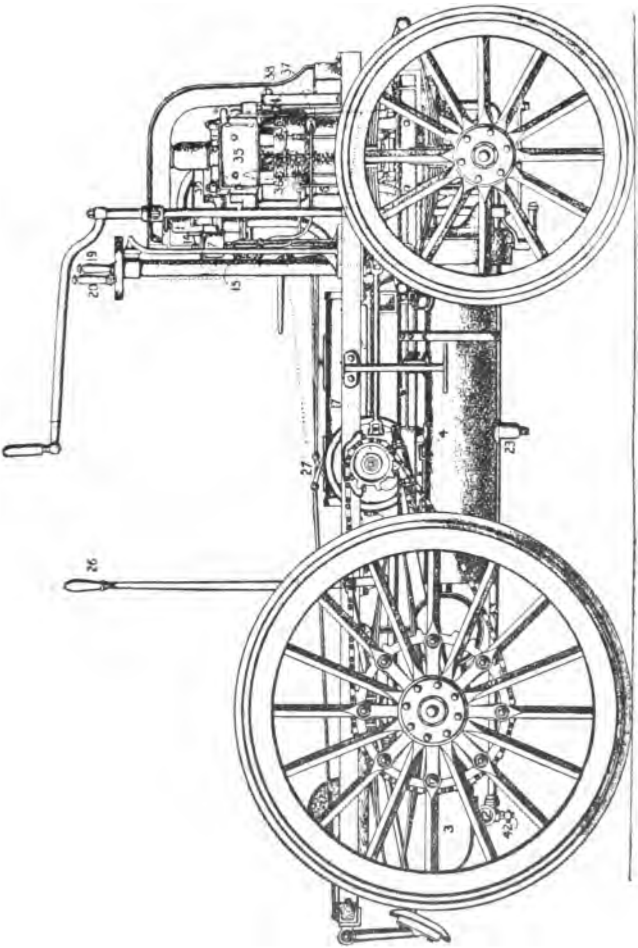


FIG. 175.

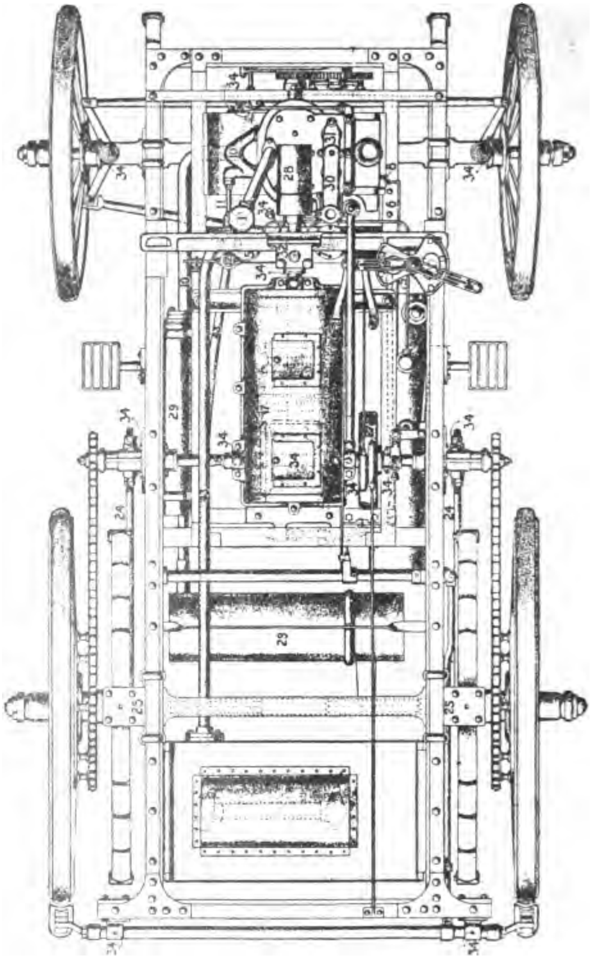


FIG. 176.

inlet valve closes automatically, and the mixture is compressed into the ignition tube at C. To insure regular ignitions, the platinum tubes should be heated in the centre to a bright red by an almost invisible flame. The tubes will require attention every three or four weeks when in constant

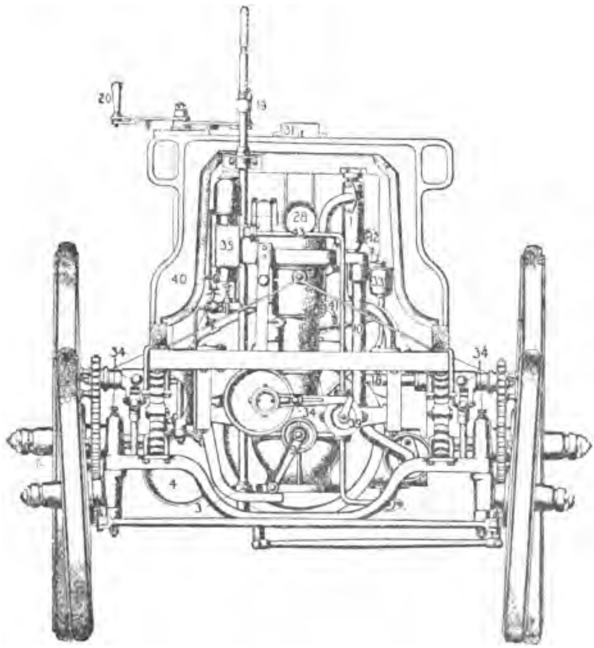


FIG. 177.

use, and should be kept perfectly clean by rubbing them lightly with the finest emery cloth. It is just as important that the tubes should be kept clean on the inside as the outside. For the removal of the tube a box key is provided, which should be carefully used, as the tubes are thin, and will not bear any rough usage. The exhaust valve is operated

by a cam. The rod attached to the valve is under the control of the governor, and is pushed aside so that the cam does not act upon it when the speed of the motor increases beyond the limit allowed. By this means the exhaust valve is held open, with the result that the products of combustion return to the cylinder, instead of a fresh charge. When the engines are in duplicate, as they always are for auto-car work, only the right-hand exhaust valve is operated first. If, however, the speed of the motor still increases, the left-hand rod is operated by the governor. This arrangement conduces to steady running. The motor, when working at full speed, makes 700 revolutions per minute. Figs. 175, 176, and 177, together with the reference to the numbers thereon, will give a detailed idea of the way in which the Daimler motor is applied to carriages.

The Daimler Motor at the Birmingham Trials.

The performances of the Daimler motor at the Birmingham trials throw light on the relative economy of the working of oil motors, and offer opportunity for comparisons between oil and steam motors. The statistics which follow are gleaned from Professor Unwin's report, printed in the journal of the Royal Agricultural Society. Professor Unwin is, however, not responsible for the deductions here made from his figures.

Looking first to the important question of cost of working the oil motor, as applied to self-moving vehicles, one cannot but notice that the Daimler motor was worked at the cost of 0·46d. per ton of cargo per mile, whereas the steam-driven motors running over the same course cost more than 0·7d. per ton of cargo per mile, representing a saving of more than thirty-three per cent, calculated on the basis of the more expensive motor. In working out these costs the coal has been taken at 20s. per ton, and the spirit at 7½d. per gallon.

Some very interesting and important figures on the resistance due to the total friction of the Daimler carriage are given. It was found that the force required to keep the

car running without acceleration on a level road was nearly 50lb. per ton. It must be noted here, however, that this figure has been ascertained by allowing the car to descend an incline with the motors revolving freely with the gearing. In the case of an oil motor the friction would not be greatly increased by the pressure exerted in the cylinders. In the case of a steam-driven motor the friction, when under steam, is likely to be increased. The reason of this is not far to seek, for the glands require to be tighter, under steam, and the resistance due to the slide valves is also increased. In the oil engine the only extra friction is that due to the side thrust occasioned by the obliquity of the connecting rods, and the lifting of the exhaust valve. Thus, it will be seen that the mechanism is credited with less frictional resistance than it might really experience, if the motor were working the car. We have said that under these circumstances of the trials it required 50lb. per ton to overcome all resistances. From other experiments on the resistance to the rolling of rubber wheels on ordinary roads, it may be estimated that about 40lb. per ton would be required to move the vehicle, if all the resistances of the mechanism were discounted. This leaves ten pounds per ton for the internal resistance of the motor mechanism, and this represents a mechanical efficiency of eighty per cent for the motor and gearing. On referring to former remarks on the efficiency of gearing, it will be understood that the design and the excellence of the workmanship must be of a high order to realise these results in practice.

The indicated horse-power required to drive a car at ten miles per hour on the level and on the various gradients is depicted in fig. 178. In calculating the points on the curve, it has been assumed that the total resistance on the level amounts to 50 lb. per ton, and the horse-powers are calculated for a gross load of one ton. From the diagram it is evident that a minimum horse-power of 1·33 is necessary for driving at ten miles per hour on good level road. Over and above this an additional power is required for propelling the

vehicle up hill. If the speed up hill be maintained at ten miles per hour, the power needed is increased very greatly. But by the use of gearing the speed up hill may be reduced, and the power accordingly reduced. How much power is required for any car depends, therefore, upon the maximum speed at which it is required to ascend the steepest gradient

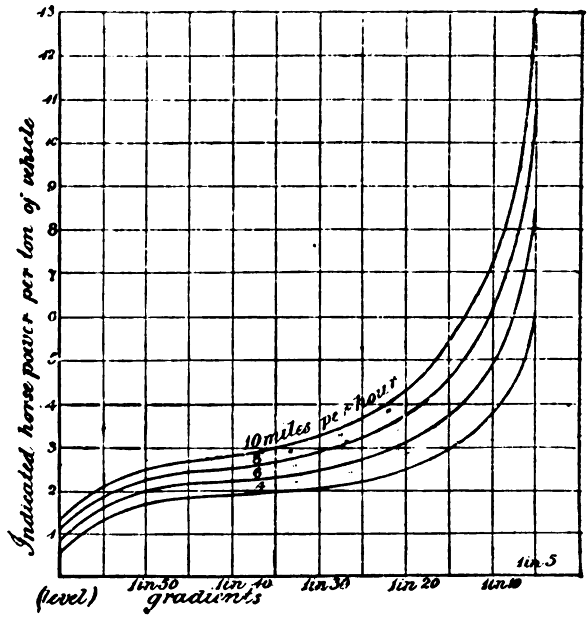


Fig. 178

that it is likely to encounter over the worst state of road. This is a matter that must be carefully considered when designing motors for autocars. It may be pointed out here, however, that the necessary horse-power required to run the vehicle at ten miles per hour on good road is doubled when an incline of one in forty-five is met, that it is trebled

when an incline of one in twenty-two is met, and that it is quadrupled when an incline of one in sixteen is met. If, in addition to the rise in level, the road surface is less favourable, it is evident that the power required will be considerably increased. It may frequently be necessary to negotiate hills of one in five, and to do this at four miles per hour a horse-power of six is needed. With such a power it will be noticed that the motor will only need to exert half its strength to mount a gradient of thirty-five to one at ten miles per hour on good roads. Although this appears wasteful, it must be borne in mind that the value of a self-propelled car depends upon its resistance to all obstacles likely to be met with. For general use, the power per ton is estimated at five indicated horse-power. Though when a car is designed specially to meet the needs of any one district, the horse-power may be considerably varied.

It is interesting to note that the post office authorities have decided to put several Daimler vehicles in service for the purpose of carrying articles transmitted through the parcels post.

Descriptive of the Drawings of the Daimler Motor.

1. Filling plug for supplying the jackets with cold water. This is to be done before starting. If in doubt as to the working of the circulating pump by which the water is made to flow freely through the jackets, this plug should be removed whilst the engine is working, and water should be seen to be passing through the pipe.

2. Perforated pipe acting as vent to the jacket water tank. This tank is filled, as above stated, until water begins to drip from the pipe 2.

3. Jacket water tank.

4. Petrol tank.

5. Pipe for admitting exhaust gases to petrol tank, to force spirit to float chamber. Pressure of gases thus admitted is controlled by valve 12.

6. Pipe from petrol tank to float chamber through which the spirit flows.

7. Float chamber.

8. Strainer through which the petrol passes on its way to the float chamber.

9. Handle for starting the motor. With ignition tubes hot and petrol up to float chamber, two or three turns of the handle should start the motor.

10. Exhaust pipe from both cylinders leading to silencer 29.

11. Pipe taking exhaust gases to petrol chamber.

12. Valve to control supply of exhaust gases to petrol chamber, acting also as relief valve in case of excessive pressure gathering in petrol chamber.

13. Strainer to prevent dirt passing to petrol chamber.

14. Spring for controlling needle valve in petrol chamber. This is always placed upon the spindle to hold down the valve when the motor is stopped for a length of time, and should be removed before attempting to start.

15. Screw for stopping motor. This acts by throwing the exhaust valve rods out of gear, thus keeping the exhaust open, and preventing the suction of a fresh charge.

16. Valve for escape of pressure from petrol chamber. This valve to be opened when stopping for any length of time.

17. Gearing box.

18. Cross shaft carrying driving pinions and bevel gears for forward or backward motion of the carriage.

19. Carriage starting lever. This may be placed in one of three positions, mid-position for stopping carriage, forward for ahead, and backward for backing carriage.

20. Lever for changing speed of carriage. Four positions possible corresponding to four speeds.

21. Clutch lever for gearing motor to driving axle 18. This must always be down before attempting to change the speed, and should be released gently to avoid jarring, after the speed lever has been operated.

22. Foot lever for operating a band brake on the driving axle.

23. Drain for letting out any water that may have collected in petrol chamber.

24. Tension rod for adjusting the driving chain.

25. Spring carrier. This to be slackened when the chain is to be adjusted.

26. Hand brake.

27. Friction drum for foot brake.

28 and G. Air induction passage. The slots sometimes require regulation, and the openings should be slightly closed when starting in cold weather.

29. Exhaust silencers.

30. Vaporiser.

31. Cover to air inlet valve to cylinder.

32. Cylinder lubricator.

33. Crank lubricator.

34. All parts with this number require to be lubricated.

35. Closed cover for burners.

36. Ignition tube burners.

37. Spring for regulating the speed of the motor.

38. Screw for adjusting above spring.

39. Circulating pump.

40. Hand pump for delivering compressed air to petrol chamber to drive petrol to float chamber before starting the motor.

41. Cylinder drain cock.

42. Drain cock to tank containing circulating water. This tank should be emptied where there is liability of frost.

43. Water circulating pipes.

CHAPTER XXVII.

Oil Engine Testing.

In a previous chapter we have described in great detail the method of testing gas engines. Much that has been said in reference thereto will readily be applied to the testing of oil engines. There are, however, some points arising in connection with oil-engine testing which require special consideration.

In the first place, it is convenient to arrange the oil supply in tanks calibrated to give the *weight* of oil. The calibration should be separately determined for each class of oil used, on account of the different density of the brands of oil now upon the market.

In testing gas engines the quantity of air per stroke is not difficult to obtain by difference, when the volume of gas in the cylinder is known. The measurement of the air supplied for combustion of the charge in an oil engine may present some difficulty, but unless this be determined the heat account cannot be ascertained. When air is supplied by a pump, an estimate of the volume may be made from the pump dimensions, though this is not satisfactory. It may, in some instances, be necessary to measure the air by arranging a suitable air trunk, and in it placing an anemometer. In the author's opinion this instrument is not reliable; nevertheless, if it be carefully tested, a correction curve may be plotted, by the application of which fairly accurate results may be obtained.

In dealing with gas-engine trials, a simple method of analysing the gases used was described. We shall now describe an equally simple method of making a determination of the carbon and hydrogen in a sample of petroleum. Although most engineers, in conducting trials, usually place samples for analysis in the hands of trained chemists, it is,

nevertheless, a great advantage, when much testing is being done, to be able to perform these simple analyses oneself.

The various petroleum oils consist chiefly, as we have previously pointed out, of carbon and hydrogen. The sum of the weights of these two substances is found to be very nearly 100 per cent. The undetermined weight may amount to 0·5 per cent, and this may be neglected in engine trial calculations. The method about to be described is due to Liebig, who made use of the fact that if the substance to be analysed be mixed with an excess of copper oxide, and the whole be heated to redness, the carbon will reduce the oxide and pass off as carbon dioxide; at the same time the hydrogen will pass off as water vapour. The carbon dioxide is absorbed by caustic potash; the water is absorbed by calcium chloride. The additional weight of these substances after absorption has taken place gives the carbon and hydrogen.

The analysis may be done in the following way: A combustion tube of hard glass about ¾ in. bore and 2 ft. long, open at one end and drawn out to a point at the other, is supported upon a light iron stand. A row of Bunsen burners is placed beneath the tube, and a sheet of asbestos arranged round the tube and the flames, in order to concentrate the latter upon the tube. Take about 0·5 gram of petroleum, and place it in a small soft glass tube, the weight of which (empty) is known. Fill the tube, seal at both ends, and carefully weigh again. Subtract from the total weight, when full, the weight of the glass tube, and so obtain the net weight of petroleum. When heavy oil is being analysed the tube need not be sealed, but in the case of light volatile oils the evaporation constantly going on at atmospheric temperature renders it impossible to determine the weight accurately unless the tube is sealed. Place the sealed tube near the closed end of the combustion tube. Then fill the remainder of the combustion tube with copper oxide.

Attach to the open end of the combustion tube a bulb

containing porous calcium chloride, so that all the gas generated passes through this bulb first. From this bulb the gas must pass into another vessel, containing a concentrated solution of caustic potash. When the weights of these vessels are ascertained, and all the joints of the tubes carefully tested, the analysis may be made. First light the burners near the open end of the combustion tube until the copper oxide becomes heated. Gradually heat the tube towards the sealed end. When the petroleum sample becomes hot the soft glass vessel in which it is contained will burst by the expansion of the liquid, and the process of reduction will commence. To prevent a violent explosion, the liquid should completely fill the sealed vessel. The products of combustion will bubble through the bulbs until the petroleum is consumed. When the gases cease to pass through, break off the sealed end of the combustion tube, and inspire the gaseous contents of the latter into the absorption vessels by means of a rubber tube.

Weigh the absorption bulbs after the process is completed, and so obtain the weights of the carbon dioxide and water. The carbon dioxide is composed of 12 parts of carbon to 32 parts of oxygen; hence $\frac{12}{44}$ or $\frac{3}{11}$ of the weight of carbon dioxide gives the weight of carbon in the petroleum sample. The weight of hydrogen will be found by multiplying the weight of water formed by $\frac{1}{9}$.

This method is satisfactory, provided you have a large excess of freshly ignited copper oxide in the combustion tube. There is, however, a liability to the formation of carbon in the combustion tube, in which case the analysis is incomplete. A modification of the above apparatus is as follows: Arrange the combustion tube with a free passage through both ends. To the end of the potash absorption bulb attach an aspirator tube, so that air or oxygen may be passed over to consume the petroleum. Whether air or oxygen be used, a calcium chloride drier should be attached to the combustion tube where the air enters, in order to absorb all moisture that might be otherwise carried over

into the absorption bulb, and credited to the hydrogen. This method has the advantage that the reduced copper may be re-oxidised by passing over a large excess of oxygen while the analysis is being carried out.

With respect to the heat given to the engine, this may be calculated from the calorific value of the oil. To determine the calorific value of oil two methods may be adopted. The determination may be made by calculation from the analysis, or by the direct combustion of a known quantity of oil in some form of calorimeter. Neither method is in itself quite satisfactory. Calculated values are often found to be higher than those obtained by calorimetric tests. On the other hand, the calorimeters at present available are not perfectly satisfactory.

To calculate the heat value of oil from the analysis—

Let C = the percentage of carbon by weight;

H = the percentage of hydrogen by weight.

Then, $14500 \{C + 4{\cdot}28\, H\}$ = *total* heat value per pound of oil.

This value includes the heat given up by cooling all the products of combustion to atmospheric temperature (60 deg. Fah.). For each pound of oil burnt about $1\frac{1}{4}$ lb. of water (according to the weight of hydrogen) is formed. If, therefore, this product passes off in the form of steam, as is inevitable in the case of internal combustion engines, the question arises, should the latent heat of the steam be deducted from the calorific value in working out the heat efficiency of the motor? The answer to the question is more a matter of opinion than of right. Both methods have been adopted in the treatment of trials. This point should, however, be referred to when reporting an engine trial, and the method adopted should be clearly stated. The author prefers to *deduct* from the calorific value of the oil the units of heat passing away with the steam.

The following example of this calculation will suffice: Upon analysis the oil yields—Carbon, 84·92 per cent; hydrogen, 15·03 per cent; undetermined, 0·05 per cent.

14,500 is taken as the calorific value of carbon, and 14,500 × 4·28 is the calorific value of hydrogen; hence the formula

$$14500\{C + 4{\cdot}28\ H\} = \text{total heat.}$$

In the example given,

$$\begin{aligned} 14500\{0{\cdot}8492 + 4{\cdot}28 \times 0{\cdot}1503\} &= \text{total heat,} \\ &= 21641 \text{ B.T.U.} \end{aligned}$$

But 1 lb. of oil will yield 9 × 0·1503 lb. of water. Each pound of water will carry away 966 units; therefore, the units to be deducted from the total value = 9 × 0·1503 × 966 = 1306.

Therefore effective calculated value

$$\begin{aligned} &= 21641 - 1306, \\ &= 20335 \text{ units.} \end{aligned}$$

With respect to the indicating of oil engines, there is great difficulty in obtaining satisfactory diagrams with some engines, more especially when the engine is being forced by the use of excessive oil supply. Hence the indicated horse power is seldom reliable, and in all cases comparisons should be made on the brake power.

After reading the description of the vaporisers used by the leading makers of oil engines, it will be noticed that all engines working with heavy oils vaporise in one of the following ways: (1) Oil is injected, together with the whole of the air supply, into a large vaporising chamber, separate from the cylinder, or (2) oil is injected into a small vaporising chamber, together with some air, but the greater volume of air enters the cylinder through a separate air valve; (3) same as (2), but all the air goes through the vaporiser; (4) oil is injected into the combustion chamber, and is vaporised therein. Air is drawn into the cylinder through a separate air valve, and mingles with the oil vapour when compressed into the combustion chamber. The first method undoubtedly possesses the advantage that the mixture of the vapour with the air is more complete.

Detracting somewhat from this, there is the fact that a large volume of highly-explosive vapour is contained in the

vaporiser, and back firing might occur with serious results. It will be observed that in this type of engine the whole of the air is heated before entering the cylinder; hence the weight of the charge is considerably reduced, and the mean pressure thereby reduced. It is sometimes suggested that difficulties in the governing are encountered by the use of this vaporiser. For if the explosions are discontinued by the cutting-out system of governing, then the vaporiser cools. Messrs. Priestman obviate this by their regulation of the oil and air supply, thus keeping the engine running even at light loads without missing fire. This method seems to be prejudicial to the economic consumption of oil at light loads. According to Professor Unwin's paper* on the Priestman oil engine, the consumption, when running with *no load*, was over 5 lb. of oil (Russolene) per hour whereas when the engine was doing nearly 7 brake horse power the oil consumption was only 3½ lb. per hour—that is, 0·988 lb. per brake horse power. Thus it appears that the method of reducing the oil supply, even though the air is also reduced, is prejudicial to economy.

From an analysis of the trials (see table of oil engine dimensions and consumptions) there appears little to choose between the various classes of vaporisers. It is important, however, to note the difference existing between vaporisers working with all the air passing through and those with only a portion. The cooling effect of the air in the former case should be carefully considered in designing the vaporiser. Results show that either methods give excellent results, as far as consumption. The mean pressure in Crossley's and Fielding's engines exceed those of other types. The mean pressure in the Hornsby-Akroyd engine (of class 4) is lower than any other. A large size of cylinder is therefore required. This is, however, compensated for by simplicity of parts and general neatness.

* Inst. C.E. Proceedings. Vol. cix. 1892.

Table of Oil Engine Dimensions and Consumptions.

Name of engine.	Cylinder sizes.			Full load indicator diagrams.			Maximum.			Oil used.	Oil consumption at full load.		Oil consumption at half load.		Method of governing.	Class of vaporiser. (See definitions in text.)	Revolutions per minute.	Authority.
	Diameter.	Stroke.	Clearance volume.	Maximum pressure.	Mean pressure	Compress'n pressure.	I.H.P.	B.H.P.	Mechanical efficiency.		Per I.H.P. hour.	B.H.P. hour.	Per I.H.P. hour.	B.H.P. hour.				
	in.	in.	c. in.	lbs. s in	bs. s. in.	lbs. s. in			Per cen		lbs.	lbs.	lbs.	lbs.			Means	
Priestman	8½	12	363	151	53·2	35	9·37	7·72	82	Royal Daylight	0·694	0·842	1·063 Russ	1·381 olene	Variable oil and air supply	No. 1	210	Professor Unwin
Hornsby-Akroyd.	10	15	638	130	28·9	65	10·3	8·57	83	Russolene	0·81	0·977	..	1·49	Variable oil supply	No. 4	239	Professor Capper
Crossley	7	15	236	238	72·2	82	7·9	7·01	88	Russolene	0·73	0·82	..	1·33	Intercepts oil supply		200	Professor Capper
Wells' "Premier"	8¼	15	360	..	49·6	..	7·3	6·46	89	Russolene	0·93	1·04	..	1·59	Exhaust kept open: Oil intercepted	No. 3	160	Professor Capper
Weyman & Hitchcock's "Trusty"	7⅝	14	..	147	44·1	38	7·04	5·98	84	Royal Daylight	0·69	0·82	0·68	1·12	Oil intercepted	No. 3	248	Mr. W. W. Beaumont M.I.C.E.
Campbell	7½	12	..	218	65·5	58	5·9	4·81	80	Russolene	0·93	1·12	..	1·30	Exhaust kept open. Oil intercepted	No. 3	208	Professor Capper
Britannia	7½	13	260	170	47	60	8·4	6·21	74	Russolene	1·25	1·63	..	1·67	Oil variable	No. 3	243	Professor Capper
Clarke-Chapman .	10½	16	..	..	..	..	..	..	..	Benzoline	..	1·25	..	1·36	Charge throttled in entering cylinder.	No. 3	..	Professor Capper
Fielding and Platt	8¼	16	..	148	79	40	..	5·28	..	Russolene	..	0·90	..	1·25	Intercept oil		220	Messrs. Fielding and Platt

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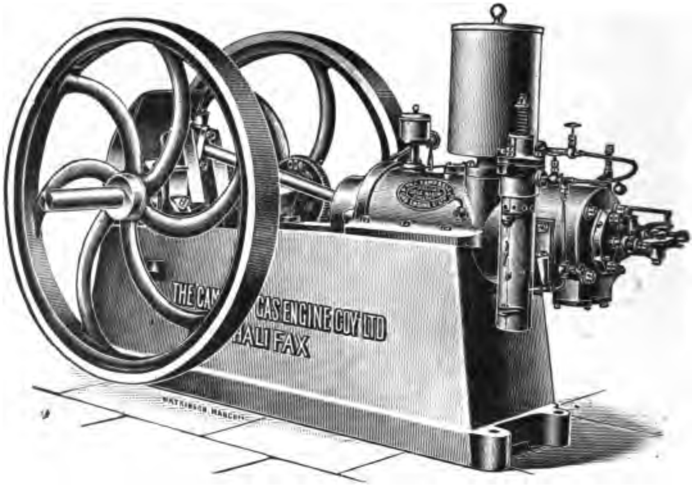

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